REPORT NO. VSE/ASG/0169-86/1ORD AD-A192 083

DTIC FILE COPY

CONCEPT DEVELOPMENT AND ANALYSIS OF THE ENVIRONMENTAL CONTROL CHEMICAL PROTECTION, AND POWER GENERATION SYSTEMS FOR THE BATTALION AID STATION AND DIVISION CLEARING STATION

Louis G. Lawrence
Michael V. McGrath
John D. Tanner
VSE Corporation
2550 Huntington Avenue
Alexandria, VA 22303-1499

DTIC ELECTE FEB 23 1988 S D H

31 March 1986

Final Report for Period 5 June 1985 - 31 March 1986

Approved for public release; distribution unlimited

Prepared for:

U.S. Army Belvoir Research, Development and Engineering Center (BRDEC)
Environmental Control Division (FE)
Fort Belvoir, VA 22060-5606

The views, opinions, and/or findings contained in this report are those of the author(s) and should not be construed as an official Department of the Army position, policy, or decision, unless so designated by other documentation.

88 2 22 203

UNCLASSIFIED
SECURITY CLASSIFICATION OF THIS PAGE

REPORT DOCUMENTATION PAGE

1a. REPORT SECURITY CLASSIFICATION Unclassified		1b. RESTRICTIVE MARKINGS -	
2a. SECURITY CLASSIFICATION AUTHORITY -		3. DISTRIBUTION/AVAILABILITY OF REPORT Approved for public release: distribution unlimited	
2b. DECLASSIFICATION/DOWNGRADING SCHEDULE -			
4. PERFORMING ORGANIZATION REPORT NUMBER(S) VSE/ASG/0169-86/10RD		5. MONITORING ORGANIZATION REPORT NUMBER(S)	
6a. NAME OF PERFORMING ORGANIZATION VSE Corporation	6b. OFFICE SYMBOL (If applicable) ASG	7a. NAME OF MONITORING ORGANIZATION Environmental Control Division (FE)	
6c. ADDRESS (City, State, and ZIP Code) 2550 Huntington Avenue Alexandria, VA 22303-1449		7b. ADDRESS (City, State, and ZIP Code) U.S. Army Belvoir Research, Development and Engineering Center (BRDEC) Fort Belvoir, VA 22060-5606	
8a. NAME OF FUNDING/SPONSORING ORGANIZATION -	8b. OFFICE SYMBOL (If applicable) FE	9. PROCUREMENT INSTRUMENT IDENTIFICATION NUMBER DAAK70-81-D-0109	
8c. ADDRESS (City, State, and ZIP Code) U.S. Army Belvoir Research, Development and Engineering Center (BRDEC) Fort Belvoir, VA 22060-5606		10. SOURCE OF FUNDING NUMBERS	

PROGRAM ELEMENT NO.	PROJECT NO.	TASK NO.	WORK UNIT ACCESSION NO.
		0193	

11. TITLE (Include Security Classification)
CONCEPT DEVELOPMENT AND ANALYSIS OF THE ENVIRONMENTAL CONTROL, CHEMICAL PROTECTION AND POWER GENERATION SYSTEMS FOR THE BATTALION AID STATION AND DIVISION CLEARING STATION

12. PERSONAL AUTHOR(S)
Lawrence, Louis G; McGrath, Michael V.; Tanner, John D.

13a. TYPE OF REPORT	13b. TIME COVERED	14. DATE OF REPORT (Year, Month, Day)	15. PAGE COUNT
Final	FROM 85/06/05 TO 86/03/31	86/03/31	

16. SUPPLEMENTARY NOTATION
-

17. COSATI CODES			18. SUBJECT TERMS (Continue on reverse if necessary and identify by block number)
FIELD	GROUP	SUB-GROUP	Battalion Aid Station Division Clearing Station M101A1 Utility Trailer

19. ABSTRACT (Continue on reverse if necessary and identify by block number) The U.S. Army Natick Research, Development and Engineering Center (NRDEC) is developing new Battalion Aid Stations (BAS) and Division Clearing Stations (DCS). A DCS will be twice the size of a BAS, as it will most likely consist of two interconnected BAS shelters. NRDEC requires that BAS shelter utilities be packaged to fit onto the U.S. Army M101A1 3/4-ton cargo trailer. The U.S. Army Belvoir Research, Development and Engineering Center (BRDEC) is developing the environmental control, chemical protection, and power generation systems (utilities) to be used with BAS/DCS shelters.

Principal conclusions and recommendations presented in this report are (1) All BAS shelter utilities can be packaged to fit onto the M101A1 cargo trailer, (2) The cargo trailer will require specific weight reduction modification to remain within the full load cross-country weight of 2840 pounds. In its new design concept the load capabilities of the M101A1 cargo trailer will not support all required DCS utilities due to increased DCS utility loads, and equipment size and weights, 4) BRDEC should proceed with the BAS utility package and trailer modification, and (5) BRDEC should further analyze and accurately determine utility loads, and equipment sizes and weights to be trailer mounted. Keywords: air inflatable portable shelters

20. DISTRIBUTION/AVAILABILITY OF ABSTRACT ☐ UNCLASSIFIED/UNLIMITED ☒ SAME AS RPT. ☐ DTIC USERS	21. ABSTRACT SECURITY CLASSIFICATION Unclassified	
22a. NAME OF RESPONSIBLE INDIVIDUAL Robert A. Rhodes, Jr.	22b. TELEPHONE (Include Area Code) (703) 664-6031	22c. OFFICE SYMBOL STRBE-FED

DD FORM 1473, 84 MAR

83 APR edition may be used until exhausted.
All other editions are obsolete.

SECURITY CLASSIFICATION OF THIS PAGE
UNCLASSIFIED

SUMMARY

The U.S. Army Natick Laboratories is developing new inflatable shelters to be used for Battalion Aid Stations (BAS) and Division Clearing Stations (DCS). These shelters are similar to those now used in the present M51 CB Pressurized POD Shelter Systems. However, the new shelters are to be approximately 50% larger by floor area.

In support of the Natick effort the U.S. Army Belvoir Research, Development and Engineering Center (BRDEC) is developing the environmental control, chemical protection, and power generation systems to be used with the BAS/DCS shelters. These new systems are packaged to fit into the U.S. Army M101A1 3/4-ton trailer, which is much smaller than the 1 1/2-ton trailer used for the M51 Shelter System.

The VSE Corporation was tasked to provide engineering evaluation and analysis, and documentation services for the BRDEC systems development effort. Part of the VSE task included using data for similar M51 systems which was provided by VSE to BRDEC under an earlier task order. Specific methods used by VSE to accomplish this task included obtaining the earlier developed M51 data for:

a. Cooling and heating loads of the present M51 CB Pressurized POD Shelter System under worldwide environmental conditions (Reference Final Report of May 1985).

b. Electrical power requirements for the present M51 CB Pressurized POD Shelter System (Reference Final Report of May 1985).

c. Electrical power requirements for the new shelter system which is scheduled to replace the M51 CB Pressurized POD Shelter System (Reference Final Report of May 1985).

d. Power requirements and recommended mechanical components to support a similarly constructed dual-walled inflatable shelter with 50% more floor area (Reference Final Report of May 1985).

Following a reevaluation of the M51 System data VSE then analyzed and evaluated the trailer mounted equipment requirements for support of the BAS/DCS units. The steps of this effort included:

a. Evaluation of each of the power components mounted on the M51 trailer for possible substitution with state-of-the-art equipment for the purpose of decreasing weight, volume and cost.

b. Preparation of a concept design for the environmental control, chemical protection, and power supply systems for the BAS and DCS. The concept addressed outdoor climate design conditions, specific power requirements, chemical protection requirements including air flow rates and pressures, allowable vehicle usage for systems, weight limitations, heating and cooling demands, human factors considerations, and dimensional constraints.

c. Evaluation of the best off-the-shelf techniques, components, processes and materials for each design.

d. Preparation of equipment/system configuration concepts mounted on the M101A1 cargo trailer, within the weight and size constraints imposed by the load handling capabilities of the trailer and associated towing vehicle.

The final product of VSE's analysis, evaluation and research is evidenced by completion of a successful concept development and the design of a 3/4-ton trailer complete with environmental control, chemical protection and power supply systems capable of supporting the BAS. However, due to the greater air conditioning requirements imposed by the larger DCS shelter, the required increased cooling capacity will result in a weight increase that could violate established trailer weight restraints. Further analysis of this concept will be performed at a later date. It should be understood clearly that the DCS, being twice the size of a single BAS, will require two utility trailers and an additional source of power generation.

PREFACE

This report was prepared at the request of the Environmental Control Division (FE), U.S. Army Belvoir Research, Development and Engineering Center (BRDEC), Fort Belvoir, VA 22060-5606. BRDEC was known as the U.S. Army Mobility Equipment Research and Development Center (MERADCOM) at the time this report effort commenced.

The concept development and analysis work described in this report was accomplished in accordance with Task Order 0169 to MERADCOM Contract DAAK70-81-D-0109. This Task Order required VSE to provide engineering evaluation and analysis, and documentation services for packaging new Battalion Aid Station (BAS) and Division Clearing Station (DCS) support equipment in the 3/4-ton M101A1 cargo trailer.

Grateful acknowledgement is given to several individuals whose help and technical input provided a valuable resource in the successful completion of this effort.

Mr. Michael Schumchyk
U.S. Army Chemical Research and Development Center
Edgewood Arsenal, MD (301) 671-4351

Major G. Flint
Academy of Health Sciences
Ft. Sam Houston, TX (512) 221-3403

Mr. Joe Boziuk
U.S. Army Natick Research and Development Center
Natick, MA (617) 651-5252

Mr. Clive Nickerson
U.S. Army Natick Research and Development Center
Natick, MA (617) 651-5252

Mr. Mark Sinofsky
Chem Fab Birdair Corporation

Mr. Danny Krimitsky
U.S. Army Research Engineering and Development Command
Ft. Belvoir, VA
(703) 664-4493

References for this report are:

a. AD-821 218L: Development of the CB Pressurized Pod System, First Quarterly Report (XM51), American Filter Co., October 1967.

b. AD-380 828L: Engineering Test of the Shelter System, Collective Protection, Chemical, Biological: XM51, Volume 1, TECOM, November 1970.

c. AD-891 870L: Technical Support to the Engineering Test of CB Shelter Systems, XM51, TECOM, December 1971.

d. Report No. 0415-19 (AD905979), Development of CB Pressurized Pod (SM51)(U), November 1972, Contract No. DAAA15-67-C-0415, awarded in March 1967 to American Air Filter Co., Inc., St. Louis, MO by Edgewood Arsenal, MD.

e. Monthly Progress Reports for Contract DAAK70-81-D-0109, Task 0169, August 1985 through December 1986.

f. McQuay O.E.M. Coil Engineering Manual, McQuay-Perfex Inc., Bernardsville, NJ, June 1978.

TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

APPENDIXES

FIGURES

TABLE OF CONTENTS (Continued)

TABLES

1. INTRODUCTION

1.1 Background. Air inflatable, pressurized shelters are used for many purposes, including medical support, by the U.S. Army. One such system is the M51 CB POD Pressurized Shelter System.

Primarily transported in a 1 1/2-ton trailer, and capable of being dropped from aircraft, the M51 is authorized for use as a Battalion Aid Station (BAS) and a Division Clearing Station (DCS). It consists of two major subsystems: entrance and shelter, and utilities trailer.

Developmental efforts are in progress to provide new BAS/DCS shelters which have 50% more floor area, and also have a new utilities trailer. The trailer selected for the new BAS/DCS shelter application is the standard U.S. Army M101A1 cargo trailer.

As part of their responsibility for the BAS/DCS shelter trailerized utility system development, the U.S. Army Belvoir Research, Development and Engineering Center (BRDEC) tasked VSE Corporation to provide engineering evaluation, analysis and documentation services for the concept development and analysis of shelter support systems, and for packaging these systems in the M101A1 cargo trailer.

1.2 Purpose of report. The purpose of this report is to document VSE's work concerned with the concept development and analysis of the environmental control, chemical protection, and power generation systems for the BAS and DCS trailerized utility system. This report was requested by the Environmental Division (FC) of BRDEC.

1.3 Scope of report. This report covers the time period of 5 June 1985 through 31 March 1986. It discusses the BAS/DCS support systems, and focuses upon packaging these systems on the M101A1 trailer. In addition to packaging support systems on the trailer, the report covers weight and balance concerns with the ultimate goal of designing a support package and trailer whose combined weight remains within the target envelope of payload and gross weight for off-the-road travel and what, if any, problems/design concerns would mandate variations of design to support DCS.

1.4 Reference to related work. Task Order 0142 to Contract DAAK70-81-D-0109 required VSE to provide engineering evaluation and documentation services in support of a utilities trailer containing support equipment for the M51 CB POD Pressurized Shelter System. Data provided to BRDEC by Task Order 0142 was the starting point of the present task. It identifies M51 power consuming components, cooling and heating loads, electrical power requirements, and more. Much of the data was used to develop new requirements for the 50% increase of BAS shelter floor area.

2. ENGINEERING ANALYSIS AND EVALUATION

2.1 Technical approach. The engineering analysis and evaluation of the Battalion Aid Station (BAS) and the Division Clearing Station (DCS) Utility System encompassed many subsystems and equipment considerations. Each

subsystem had specific parameters which had to be met in terms of functional capability, interface requirements, and weight. Extensive analysis was accomplished to determine the requirements of each subsystem and/or equipment item so as to provide proper support for the BAS and DCS shelters. Each subsystem was examined in detail with respect to required components and their interaction with each other. Also, the interface of subsystems with each other and the BAS and DCS shelters was of equal importance. As shown later, the areas of analysis and evaluation included: BAS/DCS Utility System; cargo trailer; weight analysis; human factors considerations; system operating configuration; DCS concept.

Major steps involved with performing the analysis and evaluation included:

1) Determine the type and numbers of medical equipment pertinent to the BAS/DCS mission.

2) Analyze power requirements and consumption loads for individual equipment and the BAS/DCS as total systems.

3) Determine HVAC and chemical protection requirements necessary to optimize BAS/DCS operation in worst case environments.

4) Identify the qualitative and quantitative values of equipment which nominally meet the needs of BAS/DCS mission functions and operations.

5) Conduct necessary survey, analysis and evaluation of the off-the-shelf components, materials and equipment to ascertain which, if any, would meet the established qualitative and quantitative requirements.

6) Evaluate the weight of all components and ancillary equipment, performing trade-off analyses as required to attain the design requirements of total weight not exceeding 1500 pounds.

7) Explore various design concepts to configure the total components of the power, environmental and chemical systems and necessary hardware to fit onto the M101A1 trailer.

8) Incorporate effective human factors, reliability, maintainability, safety, and standardization practices in the proposed design concept(s).

9) Develop design sketches and preliminary drawings to depict configuration of equipment/systems on the trailer.

2.2 The Division Clearing Station (DCS). The U.S. Army Natick Laboratories is in the process of developing DCS design concepts. The concept consists of two BAS's and will encompass twice the floor area of a single BAS. Each BAS will interface with its mate by use of an interconnecting airlock. The design configuration is shown in Figure 1.

2.2.1 Design considerations. In simplistic terms, the Division Clearing Station is two pressurized rib shelters interconnected to offer increased

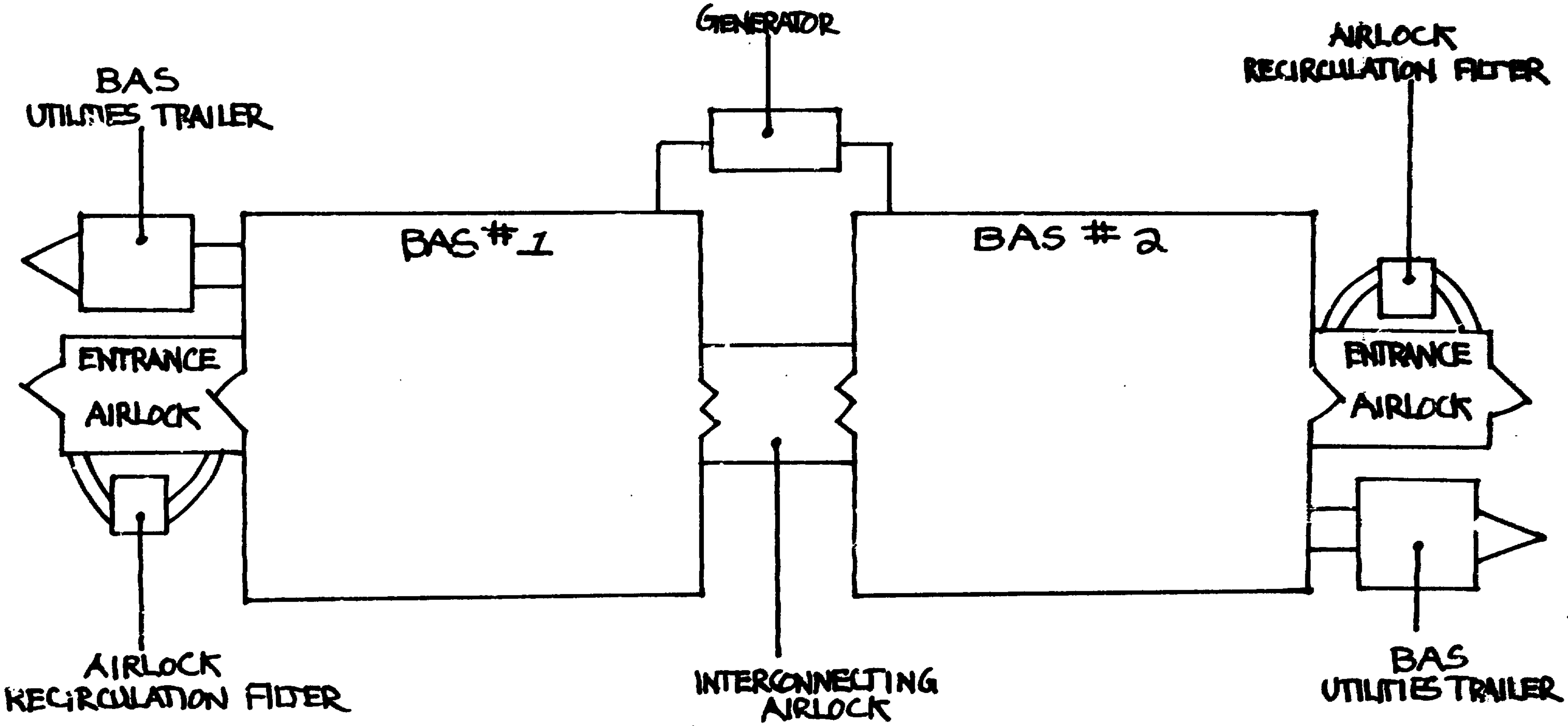


Figure 1. <u>Division Clearing Station Design Concept</u>.

floor space and an enhanced mission capability. Typically, one shelter is designated at the surgical shelter, the other as the preparatory/lab shelter. The increased mission capability requires that a substantial amount of power consuming equipment be operative in the shelters.

The surgical shelter has a connected electrical load of 9,283 watts/31,655 BTUH, and the preparatory/lab shelter has a connected load of 9,156 watts/31,222 BTUH plus an X-ray machine rated at 10,000 watts/34,100 BTUH. Should circumstances require, the total connected load to be energized at any one point in time could be 28,439 watts/96,977 BTUH electrical load imposed on the DCS utility package. This impacts the system in two respects:

1) Power must be supplied to all equipment simultaneously, and

2) The heat dissipated by this equipment must be absorbed by the trailer refrigeration system.

VSE Corporation participated in the development of EDICTS (Electrical Distribution and Illumination Components Tabulated System). During this development nine distinct medical units were identified by the Academy of Health Sciences, and a power distribution system for each medical unit was designed. The distribution system for each shelter was predicated on the assumption that the connected load was the actual load. This was not a continuous load, but due to lack of a load profile it was considered a prudent approach.

Maintenance of this approach in the DCS concept would establish the DCS electrical load at the aforementioned 28,439 watt level. This would require a 30KW generator. The applicable DOD generator for this application is the DOD Model MEP-104A, NSN 6115-00-114-1247. This generator has a dry weight of approximately 3000 pounds. Although it may not be necessary to use this particular generator, the information is helpful in making a weight assessment for application to design concept requirements. An alternative, the DOD 15KW generator DOD Model MEP 113A has a dry weight of 2500 lbs.

It is obvious that a trailer with a 1500 pound weight capacity could not transport generators of this weight. Thus, power generation and the equipment necessary to distribute this power must be provided and transported separately from the designated 3/4-ton trailer.

The result of adding power is twofold. First, it is necessary in order to supply peak load demands. Second, from a heat dissipation perspective the trailer refrigeration system must be evaluated cautiously.

As an example, the X-ray machine typically is in operation only while X-rays are required and draws rated power for the brief period necessary to effect the activity. If this period is one second, only 9.5 BTUH load would be released to the shelter. Thus, to establish the true shelter loading imposed by intermittent electrical loads, a comprehensive load profile is required. This profile can be established in two ways:

1) Develop a full scale electrical system, simulate actual operational scenarios and plot power against time or,

2) Assume theoretical performance of all pieces of equipment during various operational scenarios and plot power against time.

To be effective and valid, the effort in the first method must be accomplished by people expert in typical operational scenarios supported by personnel who are capable in electrical measurement and analysis.

To successfully accomplish the second profile requires personnel who are extremely familiar with the typical operational and performance parameters of the equipment, and personnel expert in the various operational scenarios. These persons should be augmented by electrical measurement and analysis personnel. Only at this time could a valid model of equipment operation be established, thus providing reasonably accurate estimates of potential loading. The fact is, the expertise required with respect to operation and function is not readily available.

In an attempt to identify and quantify the DCS loading requirements, much interface and discussion was accomplished with several of the individuals so noted in the preface of this report. The quantitative results of these efforts are persented in Table 1. The table identifies the quantities and types of equipment which typically are found in the DCS, and the resulting load factors expressed in BTUH. The critical information missing is the time phasing of these loads; that is, what loads are on simultaneously and for how long. Duty cycle "on time" does not provide such information.

As an example, the sterilizer in the preparatory/lab shelter may, over the period of an hour, draw 1898 watts energy. However, it can be said with certainty that at a given point in time it does not draw 1898 watts. It will either be operating and draw 5750 watts or will be non-operating and draw zero watts. Also, there is little information available to indicate whether the sterilizer will ever operate simultaneously with the sink unit and if so, for how long. If operated in tandem, the maximum load BTUH is 5,750 watts plus 1,955 watts or 7,705 total watts. If not operated in tandem, the maximum load of the two is 5,750 watts.

The length of operating time also is important. The X-ray machine, the highest peak power consumer, consumes power for such a short period of time that it is not considered a refrigeration load; yet, it is the single most significant load for the electrical generator.

The comprehensive load profile will provide this type of information. The temptation to utilize the average watts heat information and conclude that the impact of the equipment on potential heat load is 20,364 BTUH should be resisted. It is felt that this provides a low estimate of actual load, particularly in the surgical shelter during the periods of most activity. During operations, it is considered more realistic to envision that actual loads will be relatively close to connected loads. In the event of unequal loads on the two shelters, it will become necessary to mechanically induce circulation of air between shelters to maximize system efficiency.

Table 1. Division Clearing Station MED Equipment Cooling

Qty	Description (all 115V except as noted)	Amp Each	Total Amp	Total Watts	Duty Cycle on Time	Average Watts Heat
	Surgical Shelter					
1	Defibrillator	2.3	2.3	265	1.0	265
2	Sink unit, surgical	17.0	34.0	3910	0.1	390
1	Resuscitator - inhaler	0.9	0.9	104	0.5	52
2	Sterilizer, surgical	10.1	20.2	2323	0.5	1161
2	Lights, surgical field	3.0	6.0	590	0.5	345
4	Light, fluorescent, operating	2.1	8.4	966	0.5	483
1	Electro-surgical apparatus	12.1	12.1	1380	0.1	138
1	Suction unit	3.0	3.0	345	0.7	242
				9,883		3,076

3076 w x 3.41 = 10,489 BTUH

Qty	Description (all 115V except as noted)	Amp Each	Total Amp	Total Watts	Duty Cycle on Time	Average Watts Heat
	Preparatory/Lab Shelter					
1	X-Ray 100 amp 208/3/60	27.7	27.7	10,000	0.00	-
1	Sterilizer (230V)	25.0	25.0	5,750	0.33	1898
1	Sink unit, surgical	17.0	17.0	1,955	0.10	196
1	Power supply	3.3	3.3	380	0.25	95
1	Light, fluorescent, operating	2.1	2.1	242	0.50	121
10	Raiographic power, developer	0.33	3.3	380	1.00	380
1	Suction unit	3.0	3.0	345	0.70	242
1	resuscitator-inhaler	0.9	0.9	104	0.50	52
				9,156		2,984

2984 w x 3.41 = 10,175 BTUH

	Surg.	Prep/Lab	Total
Connected Load W	9,883	9,156	19,039
Diversity Load	3,076	2,984	6,060

When a representative heat load of the DCS electrical system is finally determined it must be added to the loads already imposed on the trailer refrigeration system. This will yield a total theoretical worst case-load on the refrigeration system for the BAS. A comparison of this value and the present capabilities of the refrigeration will provide an indication of what must be modified and to what extent in order to attain the required operational capabilities.

Any increases in load which exceeds the capabilities of the trailer have a pyramiding impact effect. To overcome the deficiencies, critical evaluation of the components must be accomplished. Typically, the area of greatest impact is on the evaporator and condenser coils. Increased size of the fin cross sectional areas will increase the weight of the coils. In turn, this demands a larger fan and increased envelope dimensions.

Larger fans demand larger fan motors which will create greater demands on the electrical system. This may require an increase in generator size. Also, due to greater cooling demands, the drive ratio of the compressor will probably be modified to increase compressor speed to provide greater refrigeration flow. The engine chosen to power the new utilities system, in all probability, is capable of handling the increased demands.

The bottom line once again is weight. Increased size or volume requirements most often intimate increased weight. Any addition of component weight will have to be offset by eliminating other trailer mounted components; i.e., fuel, tools or, in a last-gasp mode, sound attenuation.

Obviously, the analysis for development of a full-up 3/4-ton trailer capable of supporting DCS requirements needs to be augmented by further input of accurate, hard data developed from operational scenarios which will closely simulate those found in the field.

2.3 BAS/DCS utility system

2.3.1 Power subsystem. The BAS/DCS power subsystem consists of four major components. These components are the engine, electrical generator; mechanical drive, and refrigeration compressor.

2.3.1.1 Engine. The primary function of the BAS/DCS engine is to mechanically rotate an AC electrical generator and a refrigeration compressor. The optimum approach to the selection of the BAS/DCS engine would be to use the M51 shelter system's military standard, gasoline fueled, 20 horsepower reciprocating engine. However, because of the constraint to avoid the use of a gasoline fueled engines this could not be done.

With two exceptions, major attention was given to choosing a reciprocating engine for the BAS/DCS shelter system. These exception's were:

1) Altrudyne, a California based company who specializes in custom power system design, drew initial interest. Of particular concern was a lightweight (265 pound) 10 KW rotary generator set of small size (7.3 cu ft). Its rotary

engine is rated at 22 BHP at 3600 RPM and thus presented itself as a prime candidate. However, its multi-fuel capability did not include diesel. The local representative was contacted and in turn visited VSE. A VSE engineer explained the application, showed pictures of the power package used in the M51 system, and asked the representative to propose a diesel engine power package to include a refrigeration compressor and electric generator. Altrudyne proposed to diesel inject a foreign manufactured gasoline fueled rotary engine used in snowblower applications. The implication was that Altrudyne had some experience in modifying gasoline rotary engines to accept diesel injection. Lack of comprehensive design and performance definition, and a figure of approximately $100,000 to fully adapt this particular engine reduced confidence that this was an acceptable plan for this task. Thus, discussions with Altrudyne ceased.

2) The second concept involved gas turbine power. When a lightweight power source is required, the gas turbine is normally considered. It is usually dismissed as a suitable candidate due to its high brake-specific fuel consumption at partial loads, and its attendent high noise levels and infrared signature. For these reasons it was eliminated from further consideration.

To gain additional and comprehensive insight with respect to diesel engines, a preliminary market survey of reciprocating internal combustion diesel engines was completed in September 1985. Presented in Appendix A, the market survey report states that a review of sales literature indicated that extensive research and a detailed analysis would be required to determine acceptability of any given engine.

Two Japanese diesel engines not included in the survey and reviewed separately are manufactured by: Kubota Tractor Corporation; Yanmar Diesel Engine. Another, Isuzu Diesel of North America Inc., was included originally, but warranted further analysis. There seemed to be a number of engine characteristics common between these two manufacturer's engines which are not common to European manufacturers of small horsepower diesel engines. The Japanese engines could be characterized as high speed diesels. At the horsepower range of interest, they will operate at 3600 RPM. Typically, the weight of an engine is controlled more by the torque rating than by horsepower rating; therefore, for a given horsepower rating a high speed engine, theoretically, should weigh less than a low speed engine. The European manufacturers, for the most part, do not produce candidate engines for this application that will operate in excess of 3000 RPM. Onan is virtually the only American manufacturer of candidate engines for this application, and they disqualify themselves on the basis of weight.

All three manufacturers offered water cooled engines. All other engines surveyed were air cooled. Generally, it can be said that water cooled engines are preferred for their tendency toward a longer life expectancy.

Although water cooled engines are perceived to be heavier, inspection of the weights did not seem to verify this statistic. Noise figures were not available for any of the surveyed engines; however, it is generally conceded that water jackets mitigate combustion noises. All the air cooled engines were two cylinder engines and the water cooled engines typically had three and

four cylinders. Generally, the increased number of cylinders permits smoother engine operation, thus, generating less vibration and less structure borne noise.

The cold temperature starting requirements tended to be a high-driver consideration. Four important factors affecting cold temperature diesel engine starting include: 1) fuel characteristics, 2) induction air temperature, 3) atomization, and 4) compression ratio. Typically, the compression ratio of the European engines is 18:1. The compression ratio of the Japanese engines is 22:1 or 23:1. This greater compression ratio lends credance to their claim of good low-temperature starting and, as a result, led to further investigation of the Japanese models.

During the ensuing investigation, VSE contacted a representative of Thermo-King, and many discussions were held concerning engine drive and compressor combinations. The Thermo-King representative said that they use Isuzu diesels, and had on-going comprehensive test and evaluation programs to evaluate candidate components for their extensive mobile refrigeration applications. One such engine, the Kubota, also was of interest for the BAS/DCS research; thus, VSE explored Thermo-King's opinion of the Kubota engine. Their representative stated that the Kubota engine had been tested and, generally speaking, was rated as good. However, it was not adopted by Thermo-King because, at high temperature operation, the temperature of the engine oil was perilously close to the flash point.

From these discussions it became obvious that Thermo-King had done what few others had; evaluated a number of diesel engines in the horsepower range of interest to the BAS/DCS program. Also, they seemed to be an unbiased source of information.

During the course of the market survey Lyons & Lyons Sales Company Inc., a distributor of Ruggerini Diesel Engines, was contacted. Follow-up inquiries concerning Ruggerini diesel engines disclosed certain problems with the engines and the distributor did not recommend them for our intended application. However, the representative did recommend a Kubota, and as he was not a Kubota representative it was believed that his opinion tended to be an unbiased source of information.

At this point it was clear that the major competition was between the Isuzu and Kubota engines. Analysis of the literature did not provide a decisive edge; however, the inclination was to recommend the Isuzu because of its adoption by Thermo-King and its apparent greater logistic supportability in this country.

In order to determine the required horsepower and RPM of the selected engine it was necessary to closely examine the predicted loads of the other main parts of the power subsystem and the engine power analysis.

2.3.1.2 <u>AC electrical generator</u>. Upon comparing the generator in the existing M51 POD System with new candiate generators it was decided that replacement of the existing generator was not warranted. Also, it was

difficult to generate much interest with potential vendors as they considered it a "special item" and had difficulty justifying large design and tooling costs without a firm production commitment. The generator used currently is a two-bearing, two-pole, drip-proof, belt driven unit rated at 208 volts, 3 phase, 60 Hz, 5 KW at a power factor of 0.80 continuous duty. The generator is supplied with an appropriate voltage regulator.

2.3.1.3 Refrigeration compressor. The refrigeration compressor in the existing M51 power package is an R12 Frigidaire automotive compressor. Historically, most of the microclimate systems developed have utilized an automotive compressor. However, there remains some concern that the compressor curves presented by various manufacturers are not totally reliable due to overstatement of compressor capability. The capacity required for BAS/DCS application would require the compressor to run at speeds of approximately 4000 RPM. It is felt that continuous duty service at this speed will produce reliability problems. Keeping this in mind, if a successful utility package for the BAS could be developed and placed on a 3/4-ton trailer, the ensuing requirement would be to package a utility system for the DCS on the same trailer. Clearly, the additional capacity requirement of the DCS would put total required capacities well beyond the capability of any R12 automotive compressor running at reasonable speeds.

Analysis indicates that capacity curves of a compressor operated with chlorodifluoromethane (R22) produces approximately 60% greater capacity than with the same compressor operating with dichlorodifluoromethane (R12). Offsetting the capacity asset of R22 is the liability of approximately 50% higher condensing pressure required. However, compression ratios and horsepower-per-ton of refrigeration remain approximately the same.

Due to the emphasis on total weight and size, it would appear that the capacity advantage of R22 would more than offset the lower pressure advantage of R12. As a result, it was decided that an R22 refrigerant system would be recommended for the new system. The decision was validated further by advantages displayed using R22 evaporators and condensors. The McQuay Design Manual suggests the following capacity correction factors be applied when comparing R12 and R22 capabilities:

Evaporators: $Q_{R12} = Q_{R22} \times .9$
Condenser: $Q_{R12} = Q_{R22} \times .94$

The final selection of a compressor for application on the BAS was predicated on several variables. Of primary concern was the compressor's capability to reach demand capacities. Thermo-King's capacities are rated as follows:

- At 1500 RPM: 41,000 BTUH at 7.75 horsepower
- At 2000 RPM: 56,155 BTUH at 11.2 horsepower

Other essential capabilities include:

- Saturation suction temperature: 40°F
- Saturation discharg. temperature: 150°F
- Return gas temperature: 65°F
- Subcooling: 0°F

In addition to the capability factors, the weight of the compressor does not seem excessive; 47 pounds without service valves. Thermo-King compressors are used extensively in Thermo-King mobil commercial refrigeration systems. This application has promoted development of a compressor which could withstand the rigors of over-the-road travel and environmental exposure. This track record, in addition to the overall temperature range, seem to most closely replicate probable military applications.

Capacity and horsepower curves for the Thermo-King Model D-214 Compressor re contained in Appendix B.

2.3.1.4 Power analysis. The Isuzu diesel engine selected for this application has two loads; an AC generator and a refrigerant compressor. In addition, the engine must supply power equal to the losses of the drive system. The generator has the following loads:

		Operating Efficiency	Watts Generator Load
ECS Recirculation Fan	1 HP	75%	995
Condenser Fan	1.5 HP	75%	1,492
Ventilation Fan			400
Entrance Recirculation Fan	0.3 HP	50%	448
Lights	500 Watts	100%	500
Power Supply	100 Watts	50%	200
		TOTAL	4,035 Watts

Based on an 80% operating efficiency of the generator, the engine must supply 5044 watts or 6.76 HP to the generator input. This power is supplied to the generator through a timing belt that is 95% efficient; therefore, the total engine load imposed by generator and generator drive is 7.1 horsepower.

The refrigerant compressor utilizing R22 has the following theoretical performance at a constant saturation suction temperature of 40°F, 65°F return gas temperature and 0°F subcooling.

150°F Condensing Temperature (120°F Ambient)

RPM	Capacity BTUH	Horsepower Input
1000	27,500	4.27
1500	41,000	7.50
2000	56,000	11.15

135°F Condensing Temperature (105°F Ambient)

RPM	Capacity BTUH	Horsepower Input
1000	31,500	3.93
1500	48,000	7.10
2000	64,500	10.30

Assuming that this horsepower is supplied to the refrigerant compressor through two vee belts operating at 90% efficiency, the horsepower the engine must supply to compressor drive is as follows:

150°F Condensing Temperature (120°F Ambient)

RPM	Capacity BTUH	Horsepower Input
1000	27,500	4.74
1500	41,000	8.33
2000	56,000	12.38

135°F Condensing Temperature (105°F Ambient)

RPM	Capacity BTUH	Horsepower Input
1000	31,500	4.36
1500	48,000	7.40
2000	64,500	11.30

Derating information for the QT-23 has been received from Isuzu and is as follows:

SAE J1349	16.60 KW continuous	22.25 HP
120°F Sea Level	15.50 KW continuous	20.78 HP
107°F 500 Feet	13.50 KW continuous	18.10 HP
95°F 8000 Feet	11.90 KW continuous	15.95 HP

These values are based on prime power (continuous) rating, 3600 RPM, and gross BHP (no fan).

If five percent for fan horsepower is allowed, the following ratings occur:

SAE J1349	15.77 KW continuous	21.13 HP
120°F Sea Level	14.72 KW continuous	19.74 HP
107°F 5000 Feet	12.82 KW continuous	17.19 HP
95°F 8000 Feet	13.30 KW continuous	15.15 HP

The refrigeration curves predict a five HP compressor input requirement at a worst-case horsepower condition for the BAS at 120°F ambient (28,338 BTUH).

Therefore, the engine loading is:

Generator	7.10 HP
Refrigerant Compressor	5.00 HP
Total	12.10 HP

This is considerably less than the 120°F sea level rating of 19.74 HP. As a result, this engine was selected with thoughts of driving higher loads than those encountered in the BAS configuration.

If the generator was loaded to its rated five KW with a generating efficiency of 80% and was driven by a 95% efficient timing belt the required engine horsepower would be 8.82 HP. Theoretically, this would leave 10.92 HP to drive the refrigeration compressor. Additional refrigeration capacity would have to be accompanied by larger coils and fans thus affecting weight considerations. This information is provided to demonstrate that neither the engine nor refrigerant compressor are the limiting components in refrigeration capacity.

For ratings at altitude and high temperature, engine continuous HP decreases due to the less dense air. Simultaneously, the generator loading is expected to decrease because the less dense air will unload the fan load and heat loading will decrease due to lower temperatures encountered at altitude.

2.3.1.5 Mechanical drive system. To turn the generator and compressor at speeds which will maximize performance a mechanical, belt-driven system must be developed. Typically, the engine belt drives the electrical generator and refrigerent compressor simultaneously. However, the engine operates at 3600 RPM as does the electrical generator, but the refrigerant compressor operates effectively at approximately 2000 RPM. Thus, a speed reduction mechanism is needed to drive the refrigerant compressor. Although it is certain that the compressor will reach the required capacity at 2000 RPM during the testing phase, it may become necessary to increase or decrease this value to meet prescribed capacities for effective long term operation. The use of a belt drive system will provide a relative easy method to incrementally adjust the RPM of the refrigerant compressor if needed.

Because the engine and the generator both operate at 3600 RPM the generator could be direct driven; however, it was decided to belt drive this component also, in order to offset the opposing side-loading of the belt driven refrigerant compressor. A timing belt will be used to drive the generator. This will provide positive, non-slip engagement and increase overall generator accessibility.

To determine the optimum pulley arrangement for the compressor/generator it should be remembered that as the pulleys are moved apart the angle of contact of the drive belt on the smaller pulley will be increased. In turn, this decreases the ratio of belt tensions. The overall effect is one of decreasing drive belt tensions and placing less side-load strain on the engine

shaft. To take full advantage of this concept, the center distances of both the compressor drive and generator drive will be adjusted to minimize any imbalance placed on the engine shaft.

Although the refrigerant system will be equipped with a hot gas bypass capability for capacity control, there will be times when it will be unnecessary to operate the refrigeration system at all. The most typical example would be during operation in the heating mode. As a result, it will be necessary to include a technique which will cycle the refrigerant compressor drive on and off. This will be accomplished by including an electrically operated clutch in the compressor drive system. The clutch will be mounted on the engine shaft.

The size and weight of a clutch is proportionally closer to its torque capability than to its speed capability. Therefore, to minimize torque requirements the clutch should be installed on the shaft operating at the highest RPM. In this design, that would be the engine shaft. The pulleys should be large enough to reduce belt tension but small enough to provide reasonable belt life. In addition, loading of the engine should be accomplished as close to the flywheel mounting face as possible to minimize the moment on the engine crankshaft. The controls of the system are 28 VDC; therefore, clutch voltage should be 28 volts to preclude the requirement for two separate DC voltage sources.

2.3.2 Fuel subsystem. By design the fuel subsystem is simple and safe. Most importantly, the designated fuel to operate the BAS/DCS utilities package is diesel. This is far superior to more highly refined petrolum products such as gasoline, if for no other reason than reduced volatility.

The prime users of the diesel fuel are the derated 21 horsepower Isuzu diesel engine used to drive the refrigerant compressor and generator, and the multi-fuel capability heater integrated into the Environmental Control Subsystem (reference Section 2.3.3).

Fuel to power the diesel engine is drawn off the external trailer mounted fuel tank by the engine's internal fuel pump. Fuel at low consumption rates will vary according to the loads being placed on the equipment driven by the engine. Typically, the lower the load requirements of the generator and compressor, the lower the engine's fuel consumption. It is virtually impossible to estimate an accurate fuel consumption rate over a specified period of time. However, the advertised rate of consumption is 190 gr/ps-hr.

The multi-fuel heater also will draw its fuel from the trailer mounted tank by means of an auxiliary electric fuel pump. It is a 60,000 BTUH heater, series 10560M24B1, Stewart-Warner. The heater operates from a 23 VDC source, and employs a heated-wick ignitiion system. Fuel supply is controlled by a pulsed metering valve. Typical consumption is 0.08 lb/min HI heat, and .044 lb/min LOW heat.

Presently, the new design trailer mounted fuel tank has a capacity of 20 gallons. It will be mounted in such a way as to allow easy removal, and will be bordered by a spill tray to catch and contain any potential spillage. The configuration of the tank as shown on the design drawings (Figures 2 and 3) is rectangular. The configuration is not fixed and, in reality, may be adjusted to best suit a variety of applications. However, fuel weight as it relates to trailer balance must be considered in any final configuration design.

Fuel lines leading to the engine and heater will be hard mounted with quick disconnect, no spill type connectors. Copper lines will be used where possible with flexible lines as necessary. The fill spout for the tank has yet to be finalized in terms of configuration or position. Initially, removal of the fuel tank for refueling was the primary consideration. However, due to its weight when full, that option was eliminated. Regardless, the final fill spout design will reflect concern for ease of access and use, elimination of spillage problems and fire safety.

2.3.3 Environmental control subsystem

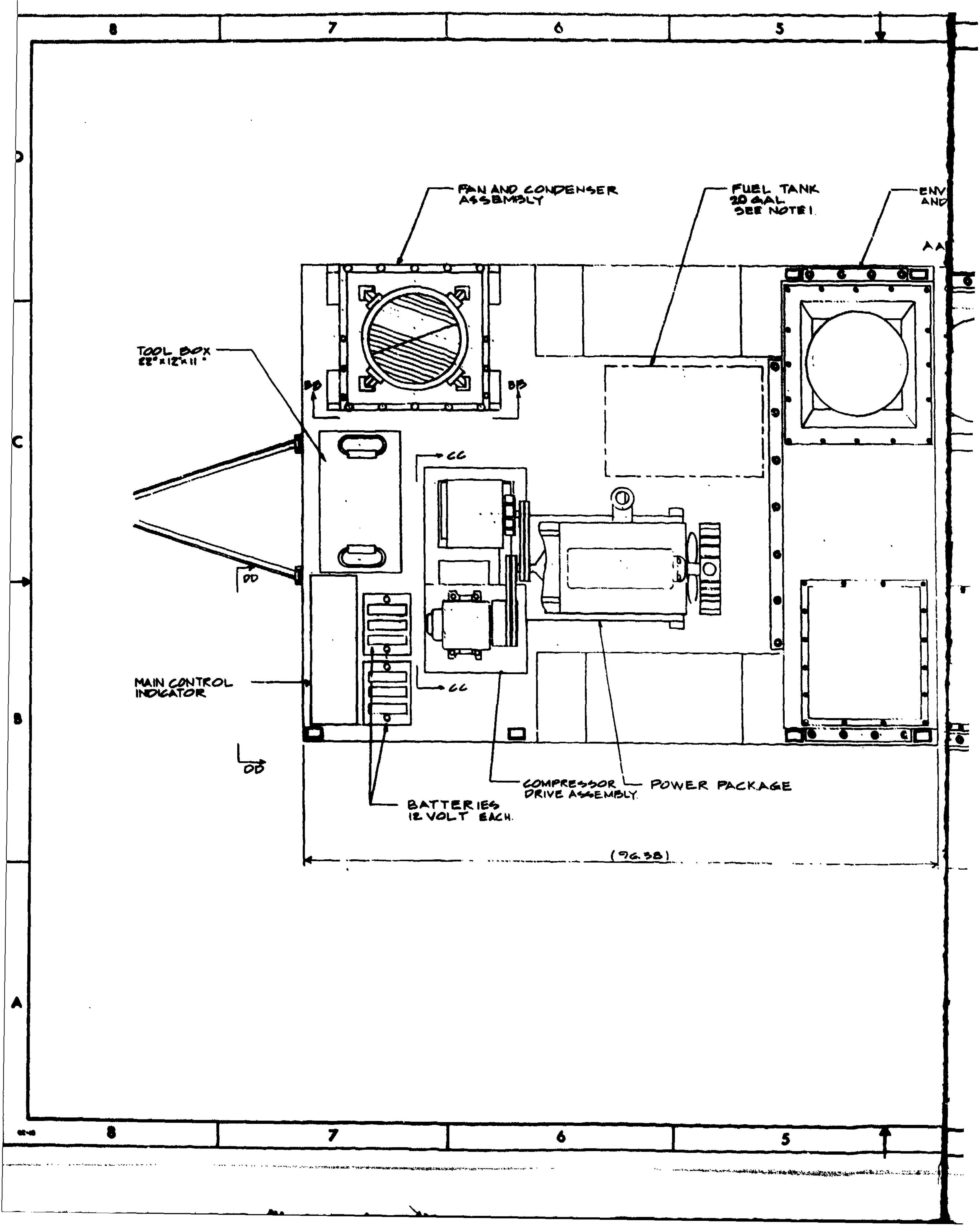
2.3.3.1 Subsystem description. The Environmental Control Subsystem (ECS) is located at the rear of the M101A1 utility trailer. The major components of the existing ECS include a recirculation fan, recirculation particulate and gas filters; evaporator and heater. These components also are included in the new concept but are augmented by the ventilation fan, ventilation particulate and gas filters and rib inflation blower within the enclosure.

The ECS enclosure will be constructed of ALUCOBOND, a composite material manufactured by Consolidated Aluminum of St. Louis. Essentially, the material is a composite rubber compound bound tightly between an outer skin of .025 thick aluminum, painted with epoxy. Typically, it is lighter than the .090 thick aluminum originally selected for ECS construction. ALUCOBOND was selected for its structural stability and its intrinsic capability to attenuate sound.

The ECS enclosure is approximately 69 inches long and is side mounted, using the full width of the trailer. It is approximately two feet wide and two feet high. Figure 2 shows the ECS mounting concept.

As viewed from the rear of the trailer there is a slight projection on the right side of the ECS to provide ventilation capability. This projection rises to a plane 43 inches above the enclosure base. On the left hand side there is a projection which provides the shelter rib inflation capability. This rises to a height of 34 inches above the enclosure base. Reference Figure 3.

In the M51 POD System, the ventilation filters occupied approximately four square feet of trailer space and the ventilation air, as well as rib inflation air, was derived from an engine driven Paxton blower. The ventilation air had to be ducted from the blower to the filters and from the filters to the environmental control enclosure. That system is more complex, heavier and



8
7
6
5
D
C
B
A
FAN AND CONDENSER ASSEMBLY
FUEL TANK 20 GAL SEE NOTE 1
ENV AND
AA
TOOL BOX 22" x 12" x 11"
BB
BB
CC
CC
DD
DD
MAIN CONTROL INDICATOR
COMPRESSOR DRIVE ASSEMBLY
POWER PACKAGE
BATTERIES 12 VOLT EACH
(96.38)

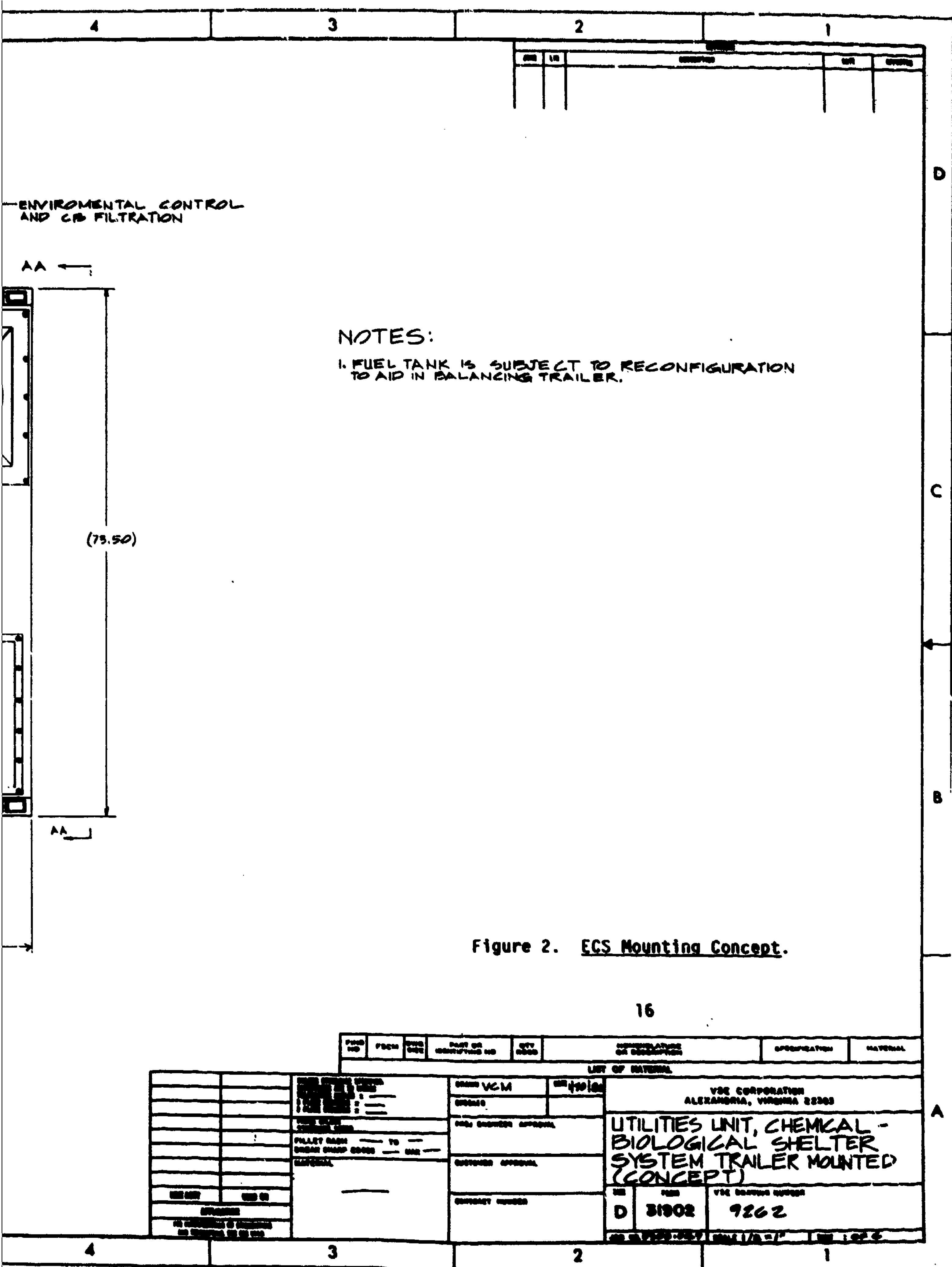
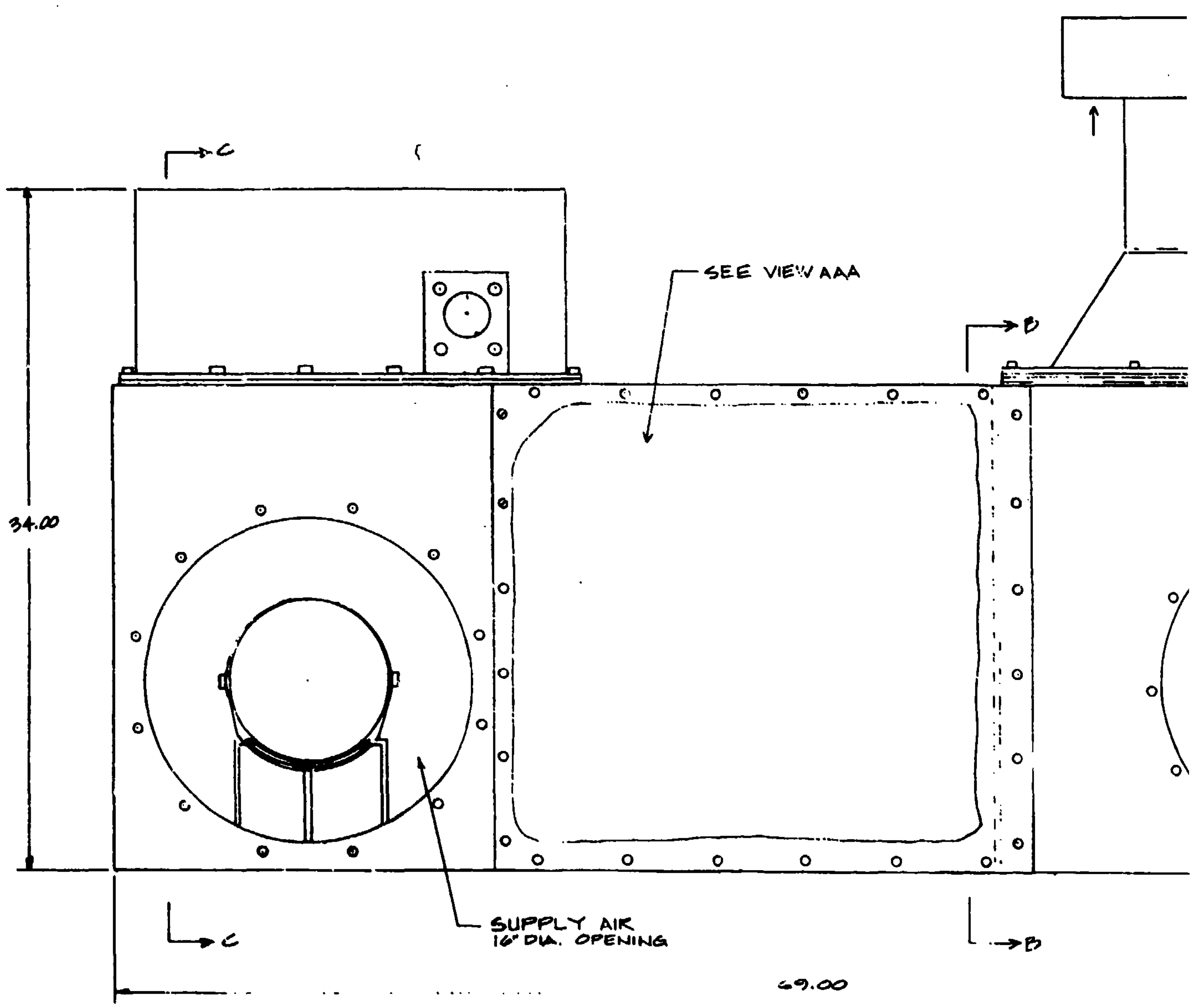


Figure 2. ECS Mounting Concept.



VIEW AA-AA
(SHOWN, ROTATED)

NOTE:
ESTIMATED WEIGHT——— 491 lbs.

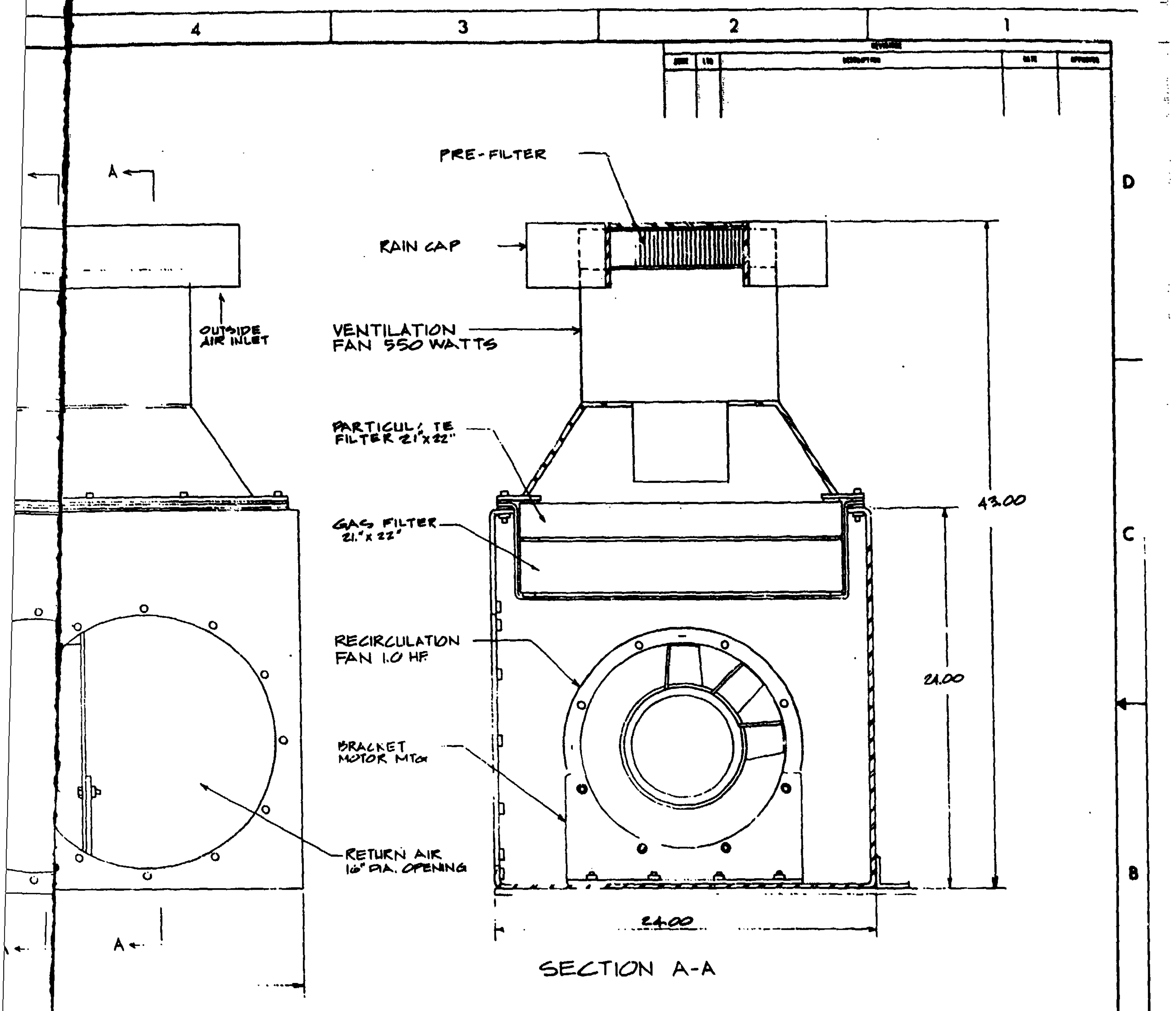
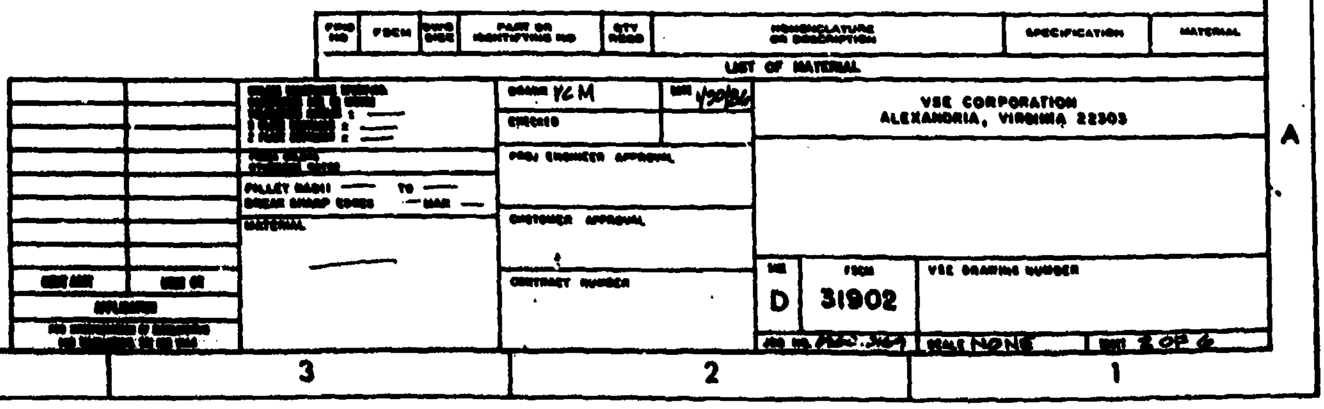


Figure 3. ECS Rear Profile.



requires greater trailer space than the proposed system. Also, greater horsepower requirements and a more complex drive system are characteristics of the existing M51 POD System, which the proposed system strives to overcome.

The M51 POD operational scenario mandates that the shelter air be circulated into and exhausted from the ECS enclosure via a 12-inch diameter flexible duct. This accounts for approximately 50% of the pressure drop of the system. Under this condition the suction side of the system is at a negative pressure as opposed to the referenced ambient pressure; therefore, any leaks on the suction side of the system potentially could contaminate the shelter during CB attacks.

The most probable solutions to this problem are twofold. The first requires an increased pressure capability for the circulation fan. To accomplish this requires an increase in the already severely limited power consumption, with a direct affect of decreased cooling capacity. The second alternative is to design the system to decrease the pressure drop now intrinsic to the present system.

At the time of this evaluation, the latter alternative seemed to be the most feasible and, as a result, the existing flexible ducts became objects of intensive analysis.

The major factors affecting pressure drops within the ducts are duct length and duct air velocity. Pressure drop is proportional to duct length and is proportional to the square of air velocity. As indicated in the System Operating Configuration section of this report (Section 2.7), duct lengths are to be kept as short as possible and free of unnecessary bends, both of which increase pressure drop.

For a given air volume, duct velocities are inversely proportional to duct area. Therefore, to decrease duct velocities we must increase duct area. Because duct areas are proportional to 0.5 x diameter^2, seemingly small changes in duct diameters can cause large changes in area. By increasing the duct size from 12 inches to 16 inches, approximately a 33% reduction in system pressure drop is realized. This, plus the shorter duct length, will virtually eliminate the possibility of the suction side of the recirculation system from contaminating the shelter during CB attack.

2.3.3.2 <u>Purging the shelter</u>. As a preliminary estimate of the time required to purge the shelter the following diffusion formula is used:

$$Q = \frac{-\ln c/co\ V}{KT}$$

Where:

c = final concentration of gas in chamber expressed in percent (%)
V = Volume of chamber (Ft^3)
K = A constant = 1
T = Purge time in minutes
co = original concentration of gas in chamber expressed in percent (%)

The sum of the end areas of the pressurized rib shelter is 344 square feet or 172 square feet per end. The length of the shelter is 22 feet, with the shelter ends tilted inward toward the center. If the ends were vertical, the volume of the shelter would be 172 square feet x 22 feet for a total of 3784 cubic feet. This represents a slightly worse-case condition than actual. Using this, the purge time is determined as follows:

$$1200 = \frac{(-\ln \frac{01}{100})\ (3784)}{1 \times T}$$

Where:

T = 21.79 minutes = 21 minutes 47 seconds

Theoretically, this will permit a 3 log reduction.

If it becomes necessary to reduce the purge time, it can be accomplished by implementing one of three possible techniques.

1) Utilize the entrance filtration system to initially purge the shelter and, upon completion, rededicate it to the entrance area. Utilizing this technique the resultant purge time is:

$$1200 + 550 = \frac{(-\ln \frac{01}{100})\ (3784)}{1 \times T} = 14.936 \text{ minutes} = 14 \text{ minutes } 56 \text{ seconds.}$$

The advantage to this method is that it can be implemented without any increase in weight or volume on the trailer.

2) Provide a separate "self-standing" filtration system to be utilized during the initial purge cycle only. This would be accompanied by full weight and volume penalty imposed on the trailer by the proposed filtration system.

3) Increase the capacity of the recirculation system which would impact significantly the size and weight of the entire Environmental Control Subsystem enclosure and refrigeration system.

While techniques 1 and 2 operate only when purging is necessary, technique 3 is in operation at all times when the system is in operation. All three techniques increase power consumption and, in turn, this occassions an increase in fuel consumption. However, techniques 1 and 2 increase fuel consumption only during the purge cycle. Technique 3 increases fuel consumption throughout the entire time the system is operating.

2.3.3.3 Cooling loads. During the scope of this effort VSE generated a report entitled "Analysis of the Heating and Cooling Loads for the M51 CB POD System and the Pressurized Rib Shelter". This final report dated May 1985 provided calculations which compared the uninsulated M51 CB POD System shelter with the pressurized rib shelter having fiberglass/wool insulating batts hung

from the shelter walls and roof. The insulating batts had an "R" value of five. It was assumed that the floor was uninsulated. As a result of the calculations it became quite evident that large heat gains during summer conditions and large heat losses during winter conditions were caused because of the uninsulated floor. To demonstrate the effect of insulating the floor a series of calculations were performed. The proposed floor was a 1" polyurethane board floor with an "R" value of 6.25. Although this information is included in the final report the values provided in the summary information is for uninsulated floors.

At the start of Task 0169 VSE was directed to use an R6 value for the pressurized rib wall and roof and an R4 to R5 for the floor. Calculations were performed with these new values and are included in Appendix C. The results are as follows:

Pressurized Rib Shelter Heat Gain

Sensible	28,338 BTUH	23,174 BTUH
Latent	0	11,633 BTUH
Total	28,338 BTUH	34,807 BTUH

2.3.3.4 Cooling load calculations. The required cooling capacity of the air conditioning system was based on two environmental conditions:

Condition 1

Outside	Inside
120°F db	80°F db
5% RH	60% RH

Ground Temp: 145°F
Solar Intensity: 355 BTUH (Sq. Ft.)

Condition 2

Outside	Inside
105°F db	80°F db
59% RH	60% RH

Ground Temp: 130°F
Solar Intensity: 343 BTUH (Sq. Ft.)

Thus, the cooling load, based on the design requirements and an outside air intake of 150 cfm is:

Condition 1

120°F outside ambient temp. sensible heat: 28,338 BTUH
Latent Heat: 0
Total: 28,338 BTUH

Condition 2

105°F outside ambient temp. sensible heat: 23,174 BTUH
Latent Heat: 11,633 BTUH
Total: 34,807 BTUH

The three major components of the refrigeration system, evaporator, compressor and condensor react to these conditions differently. Each is discussed separately:

2.3.3.4.1 Evaporator. The evaporator must have enough finned area to absorb the 1) total heat rejected by the shelter, 2) heat generated within the system and 3) ventilation air. From an intuitive standpoint it might seem that the shelter is by far the major contributor to the total. However, the data presented in Appendix C, pages C-7 and C-10, does not support this theory.

Page C-7: 120°F/80°F	BTUH Load	% of Total
Shelter Load	11,777 BTUH	41.5%
System Load Including 2,200 BTUH for people	10,686 BTUH	37.8%
Ventilation Load	5,875 BTUH	20.7%

Page C-10: 105°F/80°F		
Shelter Load	9,192 BTUH	26.4%
System Load - sensible including 2,200 BTUH for people	10,348 BTUH	
System Load - Latent including 1,800 BTUH for people	1,800 BTUH	34.9%
Ventilation Load - Sensible	3,634 BTUH	38.7%
Ventilation Load - Latent	9,833 BTUH	

For a given evaporator and evaporator temperature, the capacity varies depending on inlet conditions. When sensible heat is transferred, the driving force for transfer is the difference in dry bulb temperatures. During heat transfer between unsaturated air and a wetted surface, another factor besides the temperature difference is present. This factor is the difference in vapor pressure, which causes a transfer of mass. The transfer of mass in the proposed system occurs as condensation of water vapor from the air. Thus, the driving force for heat transfer and mass transfer are equal to the enthalpy potential.

As an example, if it is assumed that the entire evaporator surface is to be at a temperature of 40°F, the enthalpy at 80°F, 60% humidity will be approximately twice as great as at a condition of 80°F, 25% humidity. This is a good indication of the significant impact which latent loads have on evaporator ratings.

The 120°F condition is a dry coil condition and heat transfer is determined by difference in the dry bulb temperature; whereas, the 105°F

condition is a wet coil condition and heat transfer is determined by enthalpy potential.

Although the 120°F condition load is somewhat lower than the 105°F condition load, it is the 120° condition that provides the worst-case for the evaporator.

2.3.3.4.2 Compressor. Essential to understanding the refrigeration system is recognition of the fact that the entire process is not an energy conversion process but rather the transfer of heat from a low temperature source to a high temperature sink. For a given low temperature source the capacity of the compressor, with adequate condenser, will be determined primarily by the temperature of the high temperature sink. Therefore, with the given high temperature sink temperatures of 120°F and 105°F, the 120°F condition is the worst-case condition. This is verified by inspection of the compressor capacity curves in Appendix B . The compressor is impacted by total evaporator load and cannot distinguish between sensible and latent heat.

2.3.3.4.3 Condenser. The condenser receives super heated refrigerant from the compressor, removes the super heat, and then converts the refrigerant to a liquid. The condenser is the ultimate point of heat rejection from the refrigerant system. The heat rejected by the condenser is the total of heat absorbed by the evaporator and attending lines and hardware plus the heat of compression provided by the device that drives the compressor.

For design purposes it is common to assume that total compressor input power must be rejected by the condenser. The condensing temperature chosen for this system is outside ambient temperature + 30°F. This is a compromise between condenser size and compressor head pressure suggested by coil manufacturers and past military applications. Approximate condenser loads are:

	120°F	105°F
Evaporator Loads	28,338 BTUH	34,807 BTUH
Compressor HP approx.	4.3 (10,9450 BTUH	3.7 (9418 BTUH
Total	39,283 BTUH	44,225 BTUH

For the evaporator and the compressor the 120°F condition is the worst-case condition, and for the condenser the 105°F is the worst-case condition.

Evaporator capacity calculations for the 120°F conditions were made in accordance with the procedure found in the dry surface evaporator section of the McQuay O.E.M. Coil Engineering Manual, issue date June 78. Coil type 3-H, 4-row, 16 fin/inch with 420 feet per minute face velocity rates at 14,977 Btu/sq ft. A coil with finned surfaces of 20 inches x 21 inches (2.917 sq ft) will have a capacity of 43,688 BTUH. This is the size of the evaporator in the new system.

The evaporator in the M51 shelter POD System was evaluated for use in the new system. However, its finned surface of 16 x 23 3/4 inches was considered inappropriate for the size of the recirculation cabinet. McQuay was selected as the coil manufacturer because they currently are supplying coils on both the evaporators and condensers for the M51 POD Shelter System and would, therefore, be the manufacturer most familiar with the application.

McQuay has a reputation as a cost effective quality manufacturer of coils. Also, there does not appear to be a production quantity manufacturer that has a clear cut technological advantage over McQuay.

1) Refrigerant compressor. The primary function of the compressor is to recapture the refrigerant as it is converted to vapor in the evaporator and to prepare it for reuse. Preparation of the vapor consists of raising its pressure to a level corresponding to a temperature at which it can be condensed or returned to its liquid state by using an available cooling medium such as air. Therefore, the higher the temperature of the cooling medium the more the compressor must compress the vapor and the more power the compressor will consume to provide this capability.

2) Thermostatic expansion valve (TXV). The expansion valve is the dividing mechanism between the high pressure side and low pressure side of the refrigeration system. It is connected between the receiver containing the liquid refrigerant under high pressure and the evaporator in which a much lower pressure is maintained. If the expansion valve is closed, the refrigerant does not flow and consequently there is no cooling.

The expansion valve is a proportional control device. When the valve opens, the liquid in the receiver, being under pressure, is forced through the valve into the evaporator at a rate governed by the amount of valve opening. The valve monitors suction line pressure and temperature and modulates the flow of refrigerant in an attempt to maintain a constant super heat at the outlet of the evaporator. This permits the refrigeration system to respond to a wide range of evaporator heat loads.

While under pressure in the receiver, the boiling point of the refrigerant is above that corresponding to the surrounding ambient temperature and thus maintains it in liquid form. However, when it enters the evaporator, its boiling point is immediately reduced to a level corresponding to the low pressure side of the system. The temperature of the evaporator, which is higher than the new boiling point, causes the liquid to boil. As boiling occurs the vapor is drawn out of the evaporator by the compressor, and leaves room for additional liquid to enter through the expansion valve.

Selection of a thermostatic expansion valve is made on the basis of:

a) Body type. This option offers various inlet and outlet sizes and styles such as SAE flare, solder flange, pipe flange and solder.

b) Pressure drop. Subtract evaporator pressure from condensing pressure as determined during 105°F operation; add to that value friction

losses through refrigerant lines including evaporator and condenser, pressure drop across strainers, solenoid valves, hand valves and driers, pressure drop due to vertical lift of liquid line and pressure drop across refrigerant distributor. It is necessary to use the condensing pressure at 105°F operation because at a fixed orifice setting the flow of the value is determined by the pressure drop across the valve. For a fixed evaporator load, suction pressure and piping losses have a tendency to remain constant.

Valve inlet pressures are determined by condensing pressure. For a fixed design and load, condensing pressure of air cooled condensers is determined by outside ambient temperatures and air flow. For a fixed evaporator load, minimum refrigerant flow must be guaranteed at the lowest condensing pressure. This occurs at the lowest ambient temperature. At a fixed evaporator load, an increase in condensing pressure increases the flow capacity of the TXV with some degradation of performance.

c) External equalizer. The external equalizer must be used on evaporators which have refrigerant distributors. Therefore, the external equalizer connection size and style must be chosen.

d) Thermostatic charge. This is usually chosen on the basis of a manufacturer's recommendation. The information required by the manufacturer is capacity of system in BTUH, refrigerant suction temperature, condensing and liquid temperature, load temperature, type of evaporator surface and refrigerant.

Selection of the TXV valve also is dependant on the capacity of the refrigeration system and the refrigerant used in the system. Because not all necessary information is available at this time, a recommendation will not be made. However, Parker Hannifin Corp., Refrigeration and Air Conditioning Division, Alco Valve Company and Sporlan Valve Company at this point seem to be excellent sources of supply.

2.3.3.4.4 Hot gas bypass valve. Refrigerant capacity control exceeding the capacity control range of the thermostatic expansion valve may be managed by a hot gas bypass valve. As the system load requirements decrease, evaporator pressure and temperature decrease also. Without intervention this can continue until a point is reached at which the compressor will be shut off by the low pressure cut-off switch. Evaporator pressure then rises, the compressor is restarted and the cycle repeats itself. This is the most common method of capacity control for commercial refrigeration systems of five tons or less.

All military air conditioners have hot gas bypass circuits to prevent such compressor cycling. Compressor cycling is not desired because these air conditioners often are supplied with power from low KW, high impedance mobile electric generators. Air conditioners frequently are the major power consumer. When the motor driven compressor is started the current draw causes very large voltage dips in the generator voltage output. Typically, the voltage regulator attempts to correct this condition and usually results in a voltage overshoot. This oscillating condition continues until it is damped

out. The condition is not necessarily harmful to the air conditioner but may adversely effect critical power loads also provided by the generator. Not all installations in the field have critical electrical loads that would require use of a hot gas bypass circuit; however, because it is virtually impossible to predetermine which installations require or do not require it, the circuit is installed in all air conditioners.

All the factors that justify or require hot gas bypass circuits on general purpose air conditioners are not present in the M51 POD Shelter System or the newly designed system. In these systems the compressor is engine driven, and cycling the compressor does not directly effect the voltage output of the generator. Other electrical loads, fans, lights and controls are not judged to be critical in terms of minor disruptions in their power source.

It should be noted that the M51 POD Shelter System incorporates a hot gas bypass circuit. During testing it may have proven to be the most effective way to prevent evaporator frosting. The use of hot gas bypass circuits penalize refrigeration systems because the power consumption during hot gas application is unaccompanied by a concurrent refrigeration effect in the system.

In this particular application weight, as well as fuel consumption, is an important issue. Because the trailer is required to carry its own fuel for use over a specified period of operation, the operation of a hot gas bypass circuit would increase fuel consumption for an already fuel limited system.

It is suggested that a simple hot gas bypass circuit with an external equalizer and connected to a side connected distributor be installed in the new design, and a hot gas solenoid valve be installed upstream of the hot gas bypass valve. During testing, this solenoid valve may be operated to insert or remove the hot gas bypass circuit from the system. In this fashion the desirability of including the circuit in the final design can be evaluated.

The selection of a hot gas bypass valve is made on the basis of:

a) Body Type. Similar to TXV.

b) External Equalizer. The deciding factor is the amount of pressure drop between the bypass valve outlet and the compressor suction. For this design, an external equalizer is recommended. Connection types and sizes must be established.

c) Compressor capacity at minimum allowable evaporator temperature.

d) Minimum evaporator load at which the system is to be operated.

e) Condensing temperature when minimum evaporator load exists.

f) Refrigerant.

2.3.3.4.5 <u>Hot gas solenoid valves</u>. The selection of a hot gas solenoid valve involves some of the same basic items used to determine the selection of the hot gas bypass valves.

a) Refrigerant.

b) Minimum allowable evaporating temperature at the reduced load condition.

c) Hot gas bypass requirement in tons.

d) Allowable pressure drop across the valve port. For R22 a suggested value is 10 psi.

e) Coil voltage and frequency.

2.3.3.5 <u>Heating load calculations</u>. At the beginning of the present task VSE was directed to use an R6 value for the pressurized rib walls and roof, and an R4 to R5 value for the floor. Heat loss calculations were performed using these new values and are presented in Appendix D. The results tabulated on page D-4 are:

Pressurized rib shelter heat loss at -25°F is equal to 36,960 BTUH.

2.3.3.6 <u>The heating system</u>. As mentioned previously, the system heating load was calculated to be 36,960 BTUH when the outdoor ambient temperature is minus 25°F. The heater selected for this application is the Stewart-Warner 10560M24B1 series. It is an electrically controlled multi-fuel combustion heater, operated from a 24 VDC source of power and can burn DF-2, DF-A, JP-4, JP-5, and gasoline, and has a rated output of 60,000 BTUH. Each heater consists of a heated wick ignition system, two fan blowers, a burner, heat exchanger, a fuel control valve which incorporates HI and LOW heater output control, and safety controls. The heater is shrouded in a cylindrical sheet metal case with all control devices mounted for easy access. It requires an external control panel to operate the unit.

Heater models of this series are dual air source heaters. This means that although the ventilation air blowers and combustion air blowers are powered by the same motor, the design of the assembly permits individual inlets and outlets for each blower system. Because the ventilating air systems and the combustion air systems are not interconnected, variations in back-pressure imposed on one system has no effect on the other. The air needed for combustion may be piped to the heater from ambient, thus leaving the ECS free of contamination from incoming outside air or exhaust odors from the heater. This heater differs slightly from its predecessor, model series 10560M24. The main difference is the fuel control value which is replaced by, and is completely interchangeable with, the G705990 pulsed metering valve.

The heater is totally enclosed within the ECS. This maximizes the heat derived from the fuel and electrical power for transfer to the recirculated air.

This updated version of the heater used previously in the M51 POD is selected due to its satisfactory operation and the desire to use existing components of the M51 POD System for logistics, maintenance, supportability, training and standardization reasons.

2.3.4 Chemical protection subsystem

2.3.4.1 Shelter filtration. The shelter must be protected from possible chemical and biological contamination. Contamination protection is achieved by over pressurizing the shelter and by providing particulate and gas filtration. In the construction and operation of any shelter, leaks are inevitable. The pressure differential across these leaks will determine how much air will leak and in which direction it will leak. If the internal pressure of the shelter exceeds the local barometric pressure the shelter leaks out, therefore, the leaks in the chamber will not cause the ambient air to contaminate the shelter. This technique requires a replenishable source of fresh filtered air. The source of air is provided by the ventilation air circuit.

2.3.4.2 Particulate filter. The particulate filter cleanses the shelter and ECS of airborne dust and dirt. This is necessary for the protection of personnel, ECS components, and especially the gas filter and the shelter. After the initial "clean up" of the system, the primary source of particulate contamination is the dirt carried into the shelter by personnel and on equipment. Properly documented operating procedures are necessary to reduce the burden of this filter and to extend the interval between filter changes.

2.3.4.3 Gas filter. The shelter gas filter is a charcoal filter and provides a 3 log reduction of gas and aerosol contaminants.

The filters recommended for the new design are of the same type and description as the ones utilized in the M51 POD Shelter System. However, they will have to be made dimensionally different. The maximum dimensions of the face area of the M51 shelter POD system are 16 inches by 26 inches (416 sq in.) and the maximum dimensions of the filters in the new system are 21 inches x 22 inches (462 sq in.). Both sets of filters utilize flanges, and by increasing the flange dimensions identical active face areas can be achieved by maintaining the same filter thicknesses (particulate to particulate) (gas to gas). This will ensure equal filter performance.

Further information concerning these filters can be found in Appendix E.

2.3.4.4 Airlock recirculation filter. Contaminated air introduced into the entranceway airlock will be purged by recirculating the air through the airlock recirculation filter. From the standpoint of the filter, the ideal location would be inside the airlock itself. From a system design standpoint there is an interest in minimizing the volume and dimensions of the airlock to suit its primary mission; i.e., to enable two litter bearers with a litter to gain entrance into or exit from the shelter without directly exposing the shelter to the outside environment.

Inclusion of a recirculation filter which is approximately 26 inches x 27 inches x 30 inches would require an airlock of considerably greater dimensions. To minimize the dimensions, the operating location for the recirculation filter will be located outside the airlock in the contaminated

atmosphere. Airlock air is ducted to and from the filter. The proposed filter location is depicted in Figure 4. It is composed of a vanaxial fan, entrance and outlet plenums, filter holder and a particulate and gas filters.

The recirculation filter is the unit developed for the M51 POD Shelter System and represents a very efficient design for the new system. Thus, in an effort to standardize the components of the M51 System and the new design, the same recirculation filter will be employed.

2.3.5 Pressurization subsystem

2.3.5.1 Pressurized rib inflation. Inflation data for the pressurized rib shelter is:

a)	Internal volume	228 cubic feet
b)	Minimum pressure to fill ribs with air	1 psig
c)	Minimum pressure to maintain snow load	6 psig
d)	Time to inflate to 6 psig	15 minutes

The pressurized rib inflation circuit for the new design involves the use of two multi-stage centrifugal blowers operating in series. These blowers are located on top of the ECS (reference Figure 5) and draw air from the ECS at approximately ambient pressure. They are lightweight and are operational only during the short inflation period. Air is supplied through a flexible line and a check valve to the ribs. Because the maximum inflation pressure is determined by snow and wind loading that may occur on an infrequent basis, it is necessary to determine whether an adjustable or variable pressure system should be considered. This variable pressure system can be designed with predetermined discrete pressure levels. This would allow certain pressure levels to be attained for normal loading conditions and other pressure levels to be attained for snow and wind loading.

The benefit of this approach would be that the shelter rib internal pressure stresses would not exceed the levels necessary to perform effectively, thus avoiding continuous application of worst-case pressure requirements when not needed. This should have a positive impact on material life and could be implemented rather easily by modifying the control circuitry. It is suggested that this decision be made by the material developer of the shelter.

Contaminated air entering the ribs is a concern and should be addressed in terms of deployment procedures rather than equipment function. As an example, if the shelter was erected in a contaminated environment by use of the blowers alone, (without operating the ECS) the chances of contaminated air entering the ribs is good. However, if during the same scenario the ECS was being operated, the air scavenged for use by the blowers would, theoretically, be free from contamination. This issue will be addressed further as operational aspects of the system become more clearly identified.

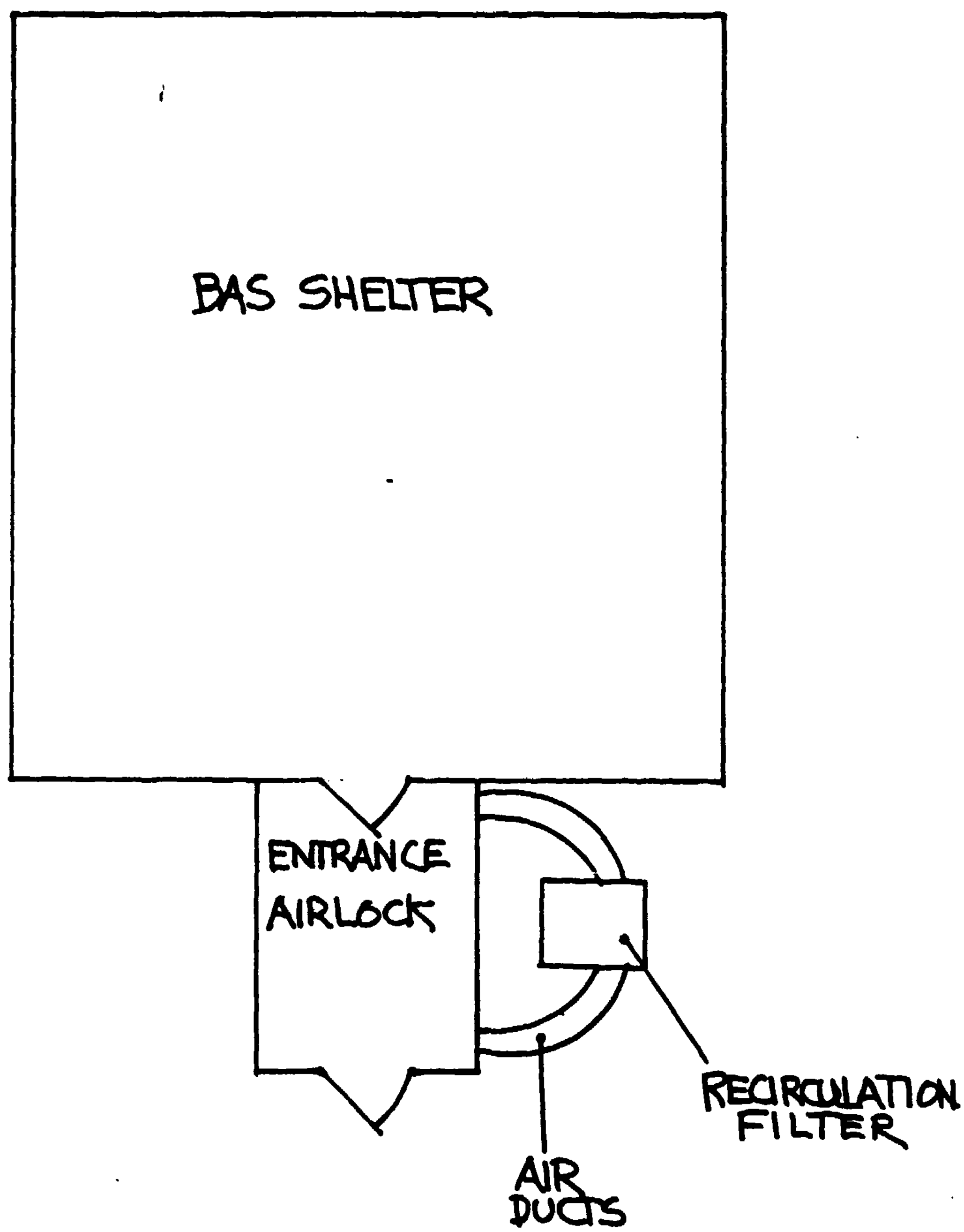
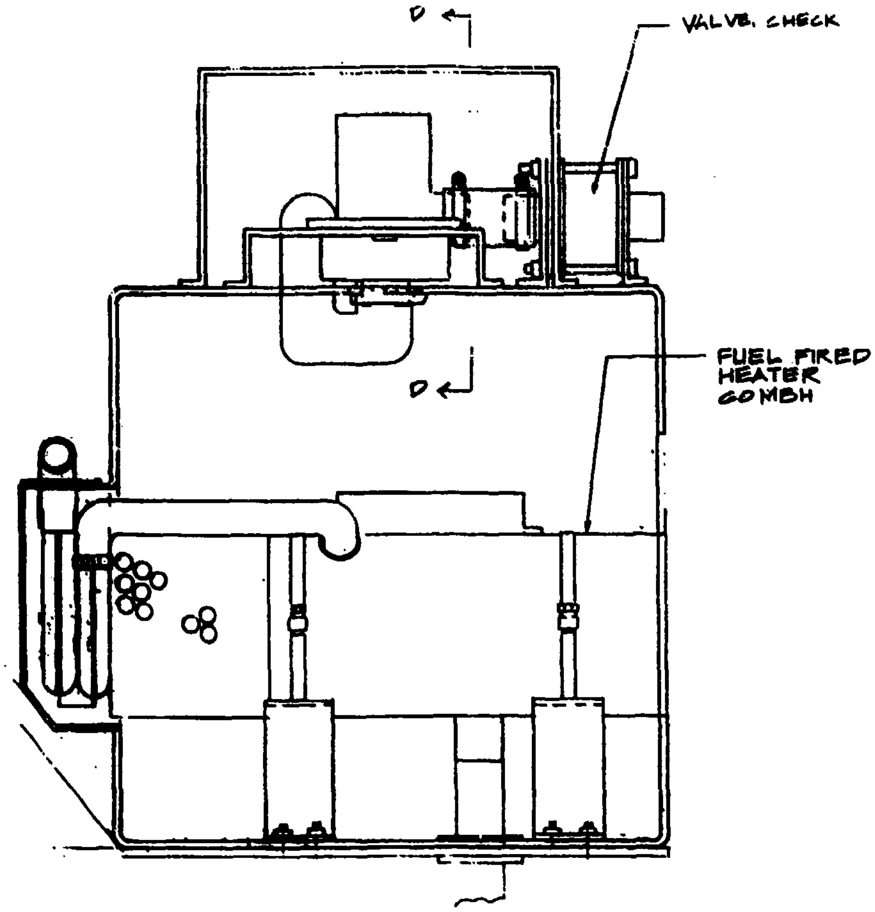
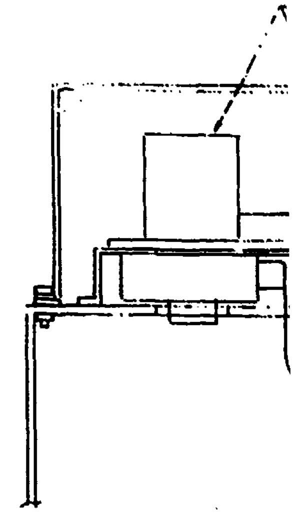


Figure 4. Airlock Recirculation Filter Configuration.

8 7 6 5

D



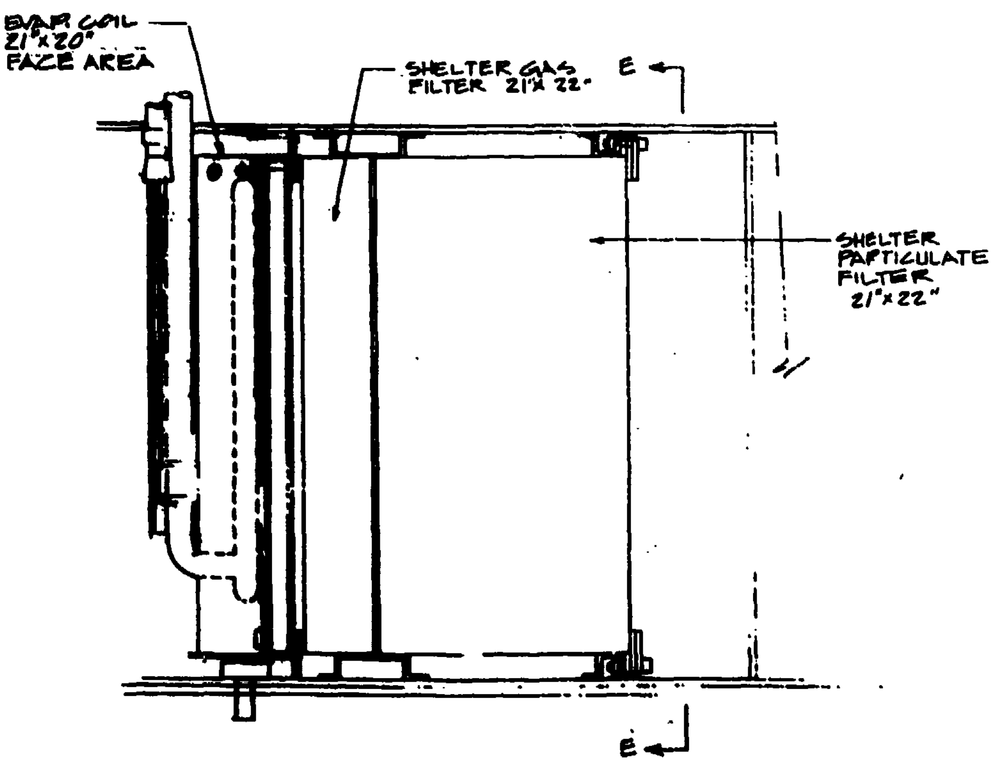


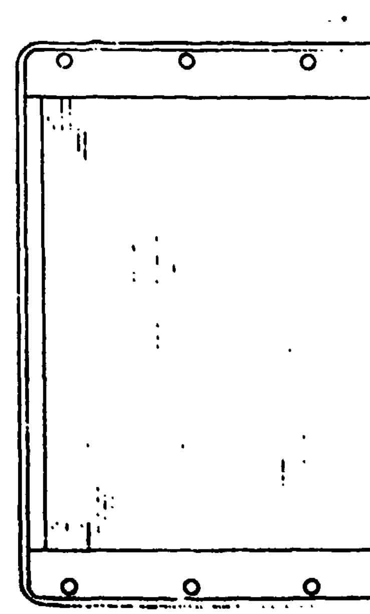
SECTIO

C

SECTION C-C

B





A

VIEW AAA

SECTION E-

8 7 6 5

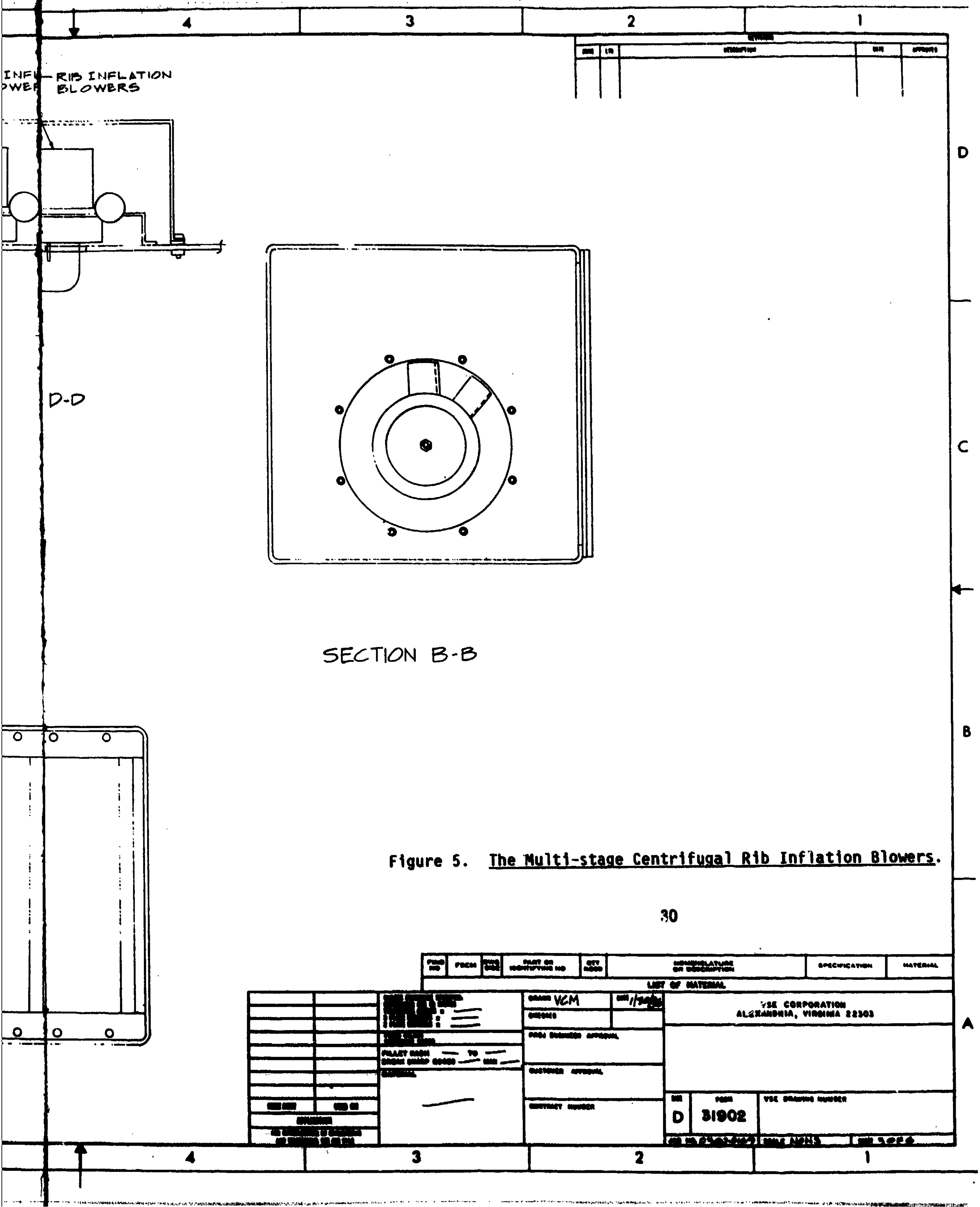


Figure 5. The Multi-stage Centrifugal Rib Inflation Blowers.

2.3.5.2 Shelter pressurization subsystem. Over-pressurization of the shelter occurs when internal shelter pressure must increase to increase air flow through shelter leaks to a flow rate equal to that which is provided by the ventilation subsystem. The ventilation subsystem is an integral part of the ECS and, as a result, the ventilation blower is required to overcome the pressure drop caused by the ECS cabinet and ducts. Because these cross-sectional areas are so large, the effect is negligible.

The ventilation system blower raises the pressure in the ECS cabinet, air ducts, and the shelter as a whole. In turn, this causes structural loads which must be met by leakage from items that, also by design, limit the achievable pressure within the shelter. The perceived value of over-pressurization of the shelter has an impact on determining shelter material loading. Snow and wind loading also impact the ultimate shelter load factor. Therefore, it is suggested that the pressure level necessary for over-pressurization should be determined by the material developer of the shelter.

2.3.5.3 System controls. Providing the capability to control the functions of various equipment of the utilities package has been addressed continuously. Obviously, some equipment needs little if any control whatsoever. Conversely, equipment such as the power package, and heating and cooling capabilities demand a certain degree of control.

At present, it is felt that the controls for certain equipment can be trailer mounted. Typically these will be of the type which are not subject to constant inspection or adjustment. However, there are some, such as the heating/cooling controls which should have the capability of being controlled from within the shelter.

Examination of the M51 POD System data package did not provide decisive information concerning the types, numbers, and functionality of all controls. This leaves ample freedom to entertain changes, but does not provide substantial evidence that the M51 controls are optimum.

Essentially, the approach deemed most reasonable at this time is to consider that controls on the trailer, and controls within the shelter are practical and feasible until proven otherwise. The proof will be evidenced by direct interface with those personnel who have thorough knowledge and/or experience with the predecessor system and the operational scenario.

2.4 The M101A1 cargo trailer. From the time it was established that the 3/4-ton M101A1 cargo trailer would be the primary trailer used to transport the BAS/DCS power facilities, it has been the subject of intense scrutiny. Primarily, the focus of attention was twofold; 1) what specific components could be put on the trailer without violating its established cross country payload limits, and 2) what could be removed from the trailer to reduce its stand alone curb weight, thus increasing payload possibilities.

For the past few months efforts to answer these questions paralleled one another. With respect to selection of lighweight, fully capable components/assemblies, special attention was paid to the cumulative total. As

it became clear that the desired utility/power capabilities could not be attained within established weight parameters, the investigation of trailer weight reduction intensified.

The M101A1 cargo trailer has a stand alone curb weight of 1340 pounds. Payload capabilities vary from a cross country payload of 1500 pounds to a highway payload of 2250 pounds. A 1500 pound payload was established as the design criterion for the utilities package, thus bringing the total gross weight to 2840 pounds.

When the sizing and weight factors of component equipment became clearly identified (Reference Section 2.5, Weight Analysis) the concern for total gross weight became paramount. As a result a discussion and evaluation as to the possibility of modifying the trailer to reduce its 1340 pounds curb weight was initiated.

To begin, a replacement of body and frame members with aluminum was considered. However, due to the mission scenario deemed most likely, replacement of the steel frame was thought to be too drastic and too tenuous in terms of frame integrity and strength. Elimination of that variable then limited any modification efforts to trailer parts above the frame.

During the course of the effort, the statement was made that replacing the steel trailer bed could possibly save some 465 pounds. This figure seemed realistic but remains unsubstantiated. However, a meeting with Mr. D. Krimitsky of Ft. Belvoir concerning a weight reduction modification of the 3/4-ton trailer provided some substantial insight.

Essentially, Mr. Krimitsky informed VSE that a prototype aluminum flatbed trailer with aluminum bows was being built at Tobyhanna. The weight savings were estimated as follows:

a) Replacing steel cargo bed with flat aluminum plate	200 pounds
b) Replaicng wood stake sides and wood bows with aluminum bows	125 pounds
Total Weight Savings	325 pounds

This seemed to be a well founded and realistic number and, therefore, modified the cross country gross payload to 1,825 pounds (1500 pounds standard plus 325 pounds of trailer modification savings). The savings ultimately proved to be very critical.

Accessibility for inspection, maintenance, and repair was a high driver. As such, it was decided that the sides as they now exist could be reduced substantially.

Focusing on that possibility produced positive results. For all practical purposes, the sides do not serve to "hold in" any equipment. All components and equipment are bolted directly to the trailer bed or are shock mounted to mounting racks/skids which bolt to the bed. This virtually eliminates the

need for high sides, including the wood sides, and the tailgate. The necessity for bows and canvas covering is appreciated, thus the only changes would be using high strength aluminum bows to support the canvas cover.

As a result of this analysis a modified trailer will be used to transport the BAS/DCS utilities package. It will include:

1) The same axle and frame structure now used with the M101A1 cargo trailer.

2) A new high strength diamond cross aluminum trailer bed. The correct thickness will be determined, and will be a function of weight vs. strength capability.

3) Replacement of existing bows with aluminum bows.

4) Mounting racks/skids for attaching components and equipment to the trailer bed. This enhances all aspects of equipment supportability. However, shock mountings will have to be appraised individually in accordance with shock tests to which the trailer would be exposed.

In summary, the modified trailer seems to be a valid design alternative. It allows the total package to reach its established design weight criterion, and provides a small amount of leeway for application in overcoming the DCS utilities demand weight problem.

2.5 Weight analysis. Weight factors have been, and remain, a high driver for the trailerized utilities package. The M51 POD Shelter System is transported on a 1 1/2-ton trailer and requires utilization of the entire 3000 pound cross-country load rating. The new system is not required to carry the entrance way (250 lbs) and the shelter itself (314 lbs); therefore, the M51 System would weigh 2436 pounds.

The new system will carry less fuel; 20 gallons of diesel fuel in lieu of 55 gallons of gasoline. This equates to an approximate 240 pound savings. A reduction of 325 pounds in the curb weight of the 3/4-ton trailer (Reference Section 2.4) manifests itself as a 325 pound savings in total payload capability. Add to this the deletion of the entranceway recirculation filter (140 lbs) and a total weight savings of 1269 pounds is afforded. This reduces the necessity of having to seek further weight savings in component selection.

Many types of components do not lend themselves to major weight reduction because their inherent technology has not yielded great tradeoffs in weight versus equivalent performance parameters. Typical components in this category are heat transfer coils, motors, fans, filters and heaters.

The requirements for a diesel engine to replace a gasoline engine has had a negative impact on total system weight. However, this is partially offset by the resulting lower fuel consumption that is responsible for the fuel savings, not to mention the safety factors.

Automotive refrigerant compressors of recent design weigh less than their predecessors. However, cooling capacities required of this system will require these compressors to be run at RPM's that would impact their long term reliability. The most negative impact is that they provide no opportunity to expand the refrigeration capability of the system to handle a DCS. However, the recommended compressor for the new design offers an increase in reliability and expansion capability which possible could handle DCS requirements but weight would increase.

Weight savings have been realized by incorporation of the rib pressurization circuit and the air ventilation system circuit into the environmental control system. Additional weight savings have been effected by lighter material construction, simplification of the power package mounting hardware, and utilization of aluminum in lieu of steel wherever and whenever possible. Table 2 presents the final individual component estimates.

It is expected that the final weights will vary somewhat; therefore, an accurate accounting of all trailer/power system weights must be kept during production to assure that established limitations are not exceeded.

2.6 Human factors engineering (HFE)

2.6.1 HFE technical approach. Application of human factors engineering to the BAS/DCS utility trailer has been a primary focus of concern. The Army requires that standards set forth in MIL-STDS 1474B and 1472C be judiciously applied to the overall system. However, many components and subcomponents of the system are commercial off-the-shelf equipment items. In so far as the individual components are concerned, the ability to apply comprehensive HFE technology is limited considerably.

As a result, the major focus of attention with respect to HFE has been how we can best fit commercial components into the overall system while assuring total HFE compatibility. To effect this a three-step approach has been employed:

STEP 1: Evaluate system components/sub-components to assure they meet what would be considered minimum HFE standards. Obviously, if there were major discrepancies the choice was to either 1) replace the component using a different make/model, or 2) examine the possibilities of modifying the component to meet HFE requirements while concurrently not voiding any applicable warranties.

STEP 2: As candidate components are identified they would be applied to the overall system configuration in the physical position thought to be the most conducive to meeting required performance standards and trailer loading calculations, particularly center-of-gravity. Once a concept arrangement has been effected, the overall configuration continues to be subjected to HFE review. The primary purpose is to assure that:

1) Any and all system/component controls are readily accessible, meet required line-of-sight, length-of-reach, and operability standards and do not offend other anthropometric restrictions.

Table 2. Final Weight Estimations for the BAS Utility System

Description	Weight, Lb.
Engine 21 HP Diesel	285
Engine Accessories	15
Generator 208/3/60, 5 KW	115
Compressors, Refrig.	47
Engine shaft extension, sheaves, belts, clutch	36
Unit skid assembly	32
Mounting brackets, supports, plates	9
Engine exhaust system	15
Batteries (2)	74
Miscellaneous hardware	30
Subtotal	658
Environmental Control and CB Filtration	
Evaporator Coil	28
Recirculating, fan, and 1 HP motor	55
Main gas filter	55
Main particulate filter	20
Make up air blower	20
Make up air gas filter	40
Make up air particulate filter	15
Prefilter	3
Tubing, fittings and valves	30
Heater, 60,000 BTUH	38
A/C controls	5
Miscellaneous	20
Subtotal	329
Sheet Metal and Miscellaneous	
Main recirculation cabinet	
Bracing, supports, frame	
Hardware	
Gaskets	
Subtotal	135
Shelter Rib Inflation System	
Rib inflation blowers	12
Hoses, tubing, fittings	10
Mounting brackets and hardware	5
Subtotal	27
Condenser Unit	
Condenser Coil	40
Condenser Fan and 1 1/2-HP Motor	60
Condenser Mounting Stand	19
Subtotal	119

Table 2. Final Weight Estimations for the BAS Utility System (Con't)

Description	Weight, Lb.
Fuel System	
Fuel tank	20
Tubing, fitting & accessories	8
Fuel	116
Subtotal	144
Electrical System	
Main power panel	30
Voltage regulator	5
RFI filter	8
Power supply 24V	35
Wiring	25
Switches Box	3
Distribution Box	4
Aux. Connection Panel	6
Miscellaneous Hardware	10
Subtotal	126
Miscellaneous	
Flexible Duct Work	35
Hoses	6
Power cables	12
Trailer modifications	50
Sound attenuation	150
Subtotal	253
Summary	
Power Unit	658
Environmental Control & CB Filtration	329
Sheet Metal and Misc.	135
Shelter Rib Inflation System	27
Condenser Unit	119
Fuel System	144
Electrical System	126
Miscellaneous	253
TOTAL	1791
Expected Weight Savings of Trailer Modifications =	-325
	1466 lbs

2) All operations, gauges, readouts, or indicators will be fully visible from a point of view outside the trailer.

3) All components requiring periodic inspection and/or maintenance activities are located in positions which will not discourage the responsible personnel from performing these tasks. It has been proven historically that difficult to perform maintenance activities do not get accomplished as required.

4) Components which have limited service life or which do fail can be replaced without having to dismantle the entire system.

5) Areas presenting possible danger to user personnel from a system safety and/or health hazard standpoint are eliminated or reduced by design.

STEP 3: As with any conceptual system, changes, modifications and rearrangements are a matter of course. Although this design appears to be semi-hardened, changes are inevitable. Each change will be subjected to thorough HFE examination to assure full understanding of potential impacts on other components, on maintainability and supportability, on operational effectiveness and most importantly, on the user personnel.

2.6.2 HFE considerations. Application of HFE to each major subsystem, component and subassembly led to significant input on overall trailer configuration. Conversely, the impact of HFE on individual components/subassemblies varied from very little to significant. The findings are presented herewith, by major component/subassembly. Figure 2 shows an overview of the total trailer configuration and location of each component/subassembly.

2.6.2.1 Fan and condenser. Of all subassemblies, it appears this is the least likely to require predetermined intermittent maintenance. Primarily, visual inspection of the subassembly is all that will be required the majority of time. Located at the forward curbside of the trailer, it can be accessed and/or removed easily should the requirement arise.

2.6.2.2 Tool box. This component will hold the majority of tools required to operate/maintain the system as a whole. The box will be water resistant and capable of removal for the purpose of user convenience. An apparent minor problem at this time seems to be the location of the box with respect to the trailer tongue frame members, but evaluation of all ramifications will continue.

2.6.2.3 Main control indicator panel. Visibility of gauges/indicators and access to control wiring play a major part in locating this component. Indicator lights and gauges will be of the size, contrast and appropriate color to be readable easily, and positioned and labeled to avoid user misunderstanding, confusion or uncertainty. Labeling will conform to MIL-STD-1473A. Proper grounding and methods for keeping the panel dry will be high-drivers.

2.6.2.4 Batteries. Batteries are located behind the Control Panel. They will be secured to the trailer for transit purposes and, although not shown on the drawing (Figure 2) cable leads will be colored red for positive and black for negative (ground). The battery terminals will be marked clearly, "POS" and "NEG" respectively. The possibility of using a battery box is being evaluated.

2.6.2.5 Compressor/generator. These individual components will be mounted on a rack/frame which, in turn, will be shock mounted onto the trailer bed. At this time, drive assemblies are perceived to be belt drives. This warranted attention to replacement capability and, thus, accessibility. As configured, there is ample room to maneuver around the subassembly and, in fact, the compressor may be reached from the streetside of the trailer. All electrical connectors will be labeled and keyed to the greatest extent possible to avoid inadvertent misconnection. Belt guards will be required.

2.6.2.6 Engine. As one of the heaviest components, particular attention was given to the location of the engine. Also, the requirement for frequent inspection and maintenance was a major consideration. The results of these variables have placed the diesel engine slightly to the streetside of the trailer bed center. This may have to be adjusted nominally to accomodate unrestricted access to coolant fill and overflow areas, oil, air and fuel filters, oil drain plugs, various gauges etc. However, at this time every indication is that the engine remains highly accessible. It will be shock mounted on a frame rack/skid.

2.6.2.7 Environmental control subsystem. As configured presently the ECS is mounted on the very rear (tailgate) section of the trailer, requiring the entire area from streetside to curbside. Of the various subassemblies which comprise the ECS, the one which demands the greatest accessibility is that containing the gas and particulate filters. Referring to Figure 3, the gas and particulate filter area is located in the center of the ECS. This design was effected to enhance accessibility to the filters for obvious reasons. As designed, the filter cover will be totally removeable, using captive hardware, thus allowing unrestricted access to the filters. Preliminary height above ground analysis indicates that the trailer bed is approximately 35 inches above ground level. The ECS rises approximately 24 inches above that level, thus placing the filter access area between 35 inches and 59 inches above ground. For maintenance purposes this will accomodate even the 5th percentile personnel quite well.

It is understood that filter replacement may have to be performed by personnel wearing protective gear; as such, retaining hardware and removal/replacement procedures will be designed to enhance the process.

2.6.2.8 Fuel tank. The fuel tank presently is located to the curbside of the diesel engine. Its configuration is very flexible but remains subject to proper weight distribution. The capability to fill the fuel tank and read its fuel level has been given consideration but remains subject to final design. A drip/catch well will surround the tank to contain any fuel

spillage/leakage. The tank should be removeable and well labeled for indicating the type of fuel to be used and any pertinent refueling instructions.

Obviously, HFE ramifications are many and varied. Major impacts and inputs have been thoroughly analyzed. However, as the utility system concept hardens, the HFE impacts will vary in scope and intensity, thus providing an iterative design and feedback process. The results will manifest themselves through proactive consideration of design, not retrofit of the product.

2.6.3 Noise levels and noise attenuation

2.6.3.1 Noise levels. Due to the anticipated position of the BAS/DCS utility system to the BAS/DCS shelters, not to mention the adamant stance taken by the Surgeon General and Army Managing Activities, any noise produced by the utility system must be reduced to levels stipulated in MIL-STD-1474B.

To start, it is difficult to anticipate noise levels on a system which has not been built. Quantifiable measurements from predecessor systems may provide some indication of the level of effort required, but introduction of new assemblies and components make previous noise data all but invalid. As such, we have taken the approach of designing noise attenuation into the system instead of trying to attenuate the noise once the design is hardened. The following paragraphs describe what has been done to date. This is by no means a doctrine to which we are inseparably joined, rather it represents initial recognition of potential noise problems and ensuing attempts to reduce their impact. No doubt, as the concept continues to develop other noise origins will surface, thus forcing continued analysis and evaluation.

2.6.3.2 Noise attenuation. At this time there are clearly two major categories of noise with which we must contend; 1) air-borne noise, and 2) structure-borne noise. Both of these will be present with the trailer. Unfortunately, the two often are caused by the same equipment, thus requiring a combination of abatement methods.

Essentially, any rotating equipment will produce noise. This implies that the various fans, compressors, generators, inflation blowers, the engine, and the timing and belt driven components are going to produce noise. This seems ominous at first until you recognize that one component's noise could easily overwhelm all others cumulatively. Typically, the engine is the culprit. It will be no different in this case. However, quieting the engine noise has the unsettling effect of making other noises noticeable. The question then is not which component should receive primary attenuation emphasis, but rather, how can we best quieten all noise producers. This is the approach we have chosen to use.

To accomplish satisfactory noise attenuation, components which lend themselves to produce structure-borne noise will be shock mounted and, if applicable, their mounting frames/skids will be shock mounted to the trailer. Also, internally mounted fans such as the condenser and ventilation fan will be soft mounted, if design permits.

The primary noise source is the diesel engine and its ancillary components, the compressor and generator. At this point they are to be totally encapsulated with a protective sound absorbtion material. We anticipate using ALCUBOND (discussed in Section 2.3.3.1) with a sound absorbitive material (to be determined) attached to the interior. In addition, exhaust silencing will be accomplished using a well-designed muffler. Of course engine heat dissipation will be a high driver in any design efforts.

From this point, noise attenuation becomes a matter of seeking and isolating the noise source, then devising the best methods to attenuate the sound. Efforts to accomplish this, and to meet the established dBA levels will remain a high priority.

2.7 System operating configuration. To assure to the greatest extent possible that the concept design will perform to expected capabilities, much evaluation and analysis was required. Equally important however, was the evaluation of how and where the design will fit into an operational scenario. Consideration of these factors, and continuous reevaluation of their importance typically produces an accurate system operating configuration. To a great extent this has been accomplished with the new BAS/DCS design effort.

Examination of the previous operational configuration, shown in Figure 6, shows the BAS with the air-lock entrance on one end, complete with an external recirculation filter, and the utilities trailer placed at right angles to the shelter. The required supply and return duct lengths to complete the right angle turn and enter/exit the shelter was approximately 16 feet. This had both a positive and negative effect.

The positive effect involved noise attenuation. In reality, the further the trailer is located from the shelter the less the noise impact on shelter occupants. The negative impacts were a large length of duct with which airflow has to contend, the decreased capability to reach desired airflow characteristics and additional storage and wieght parameters.

2.7.1 Trailer location. The proposed configuration for the new design, shown in Figure 7, places the utility trailer parallel to the airlock entrance way. The impacts of such a configuration were weighed carefully prior to concluding that it was the most acceptable.

Impacting heavily on the evaluation of proposed configuration was the resultant elimination of 50% of duct length. The environmental control unit is positioned on the rear of the trailer, and effectively faces the shelter, thus creating a straight-in approach to ductwork. The weight, storage and air supply power requirements benefit from this configuration as well.

Noise attenuation, however, is a major problem. It has been addressed in terms of effectively reducing the noise by applying sound attenuation technology, not by moving the trailer further away. The results here should prove to be an impressive improvement.

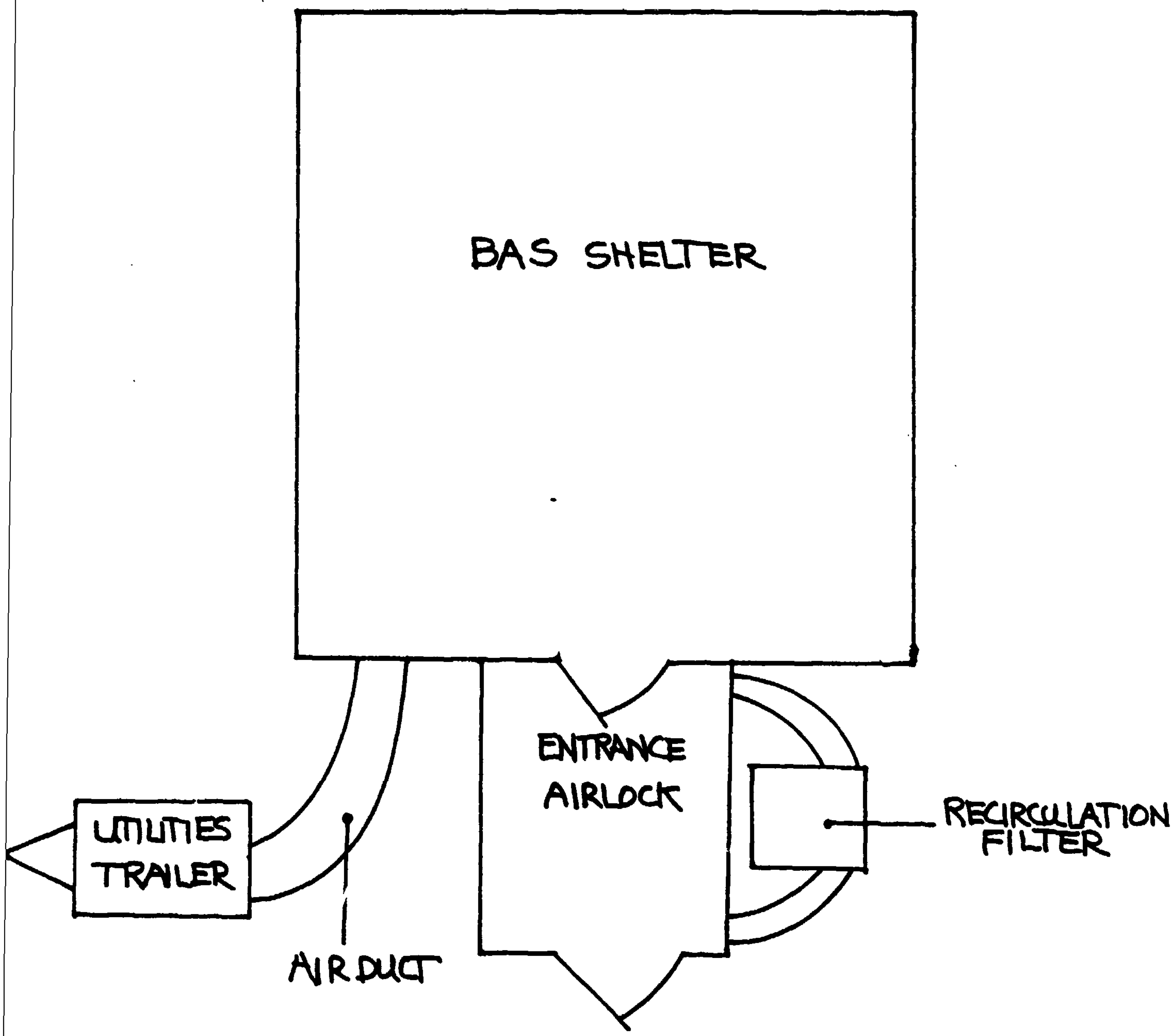


Figure 6. **BAS - Previous Operational Configuration.**

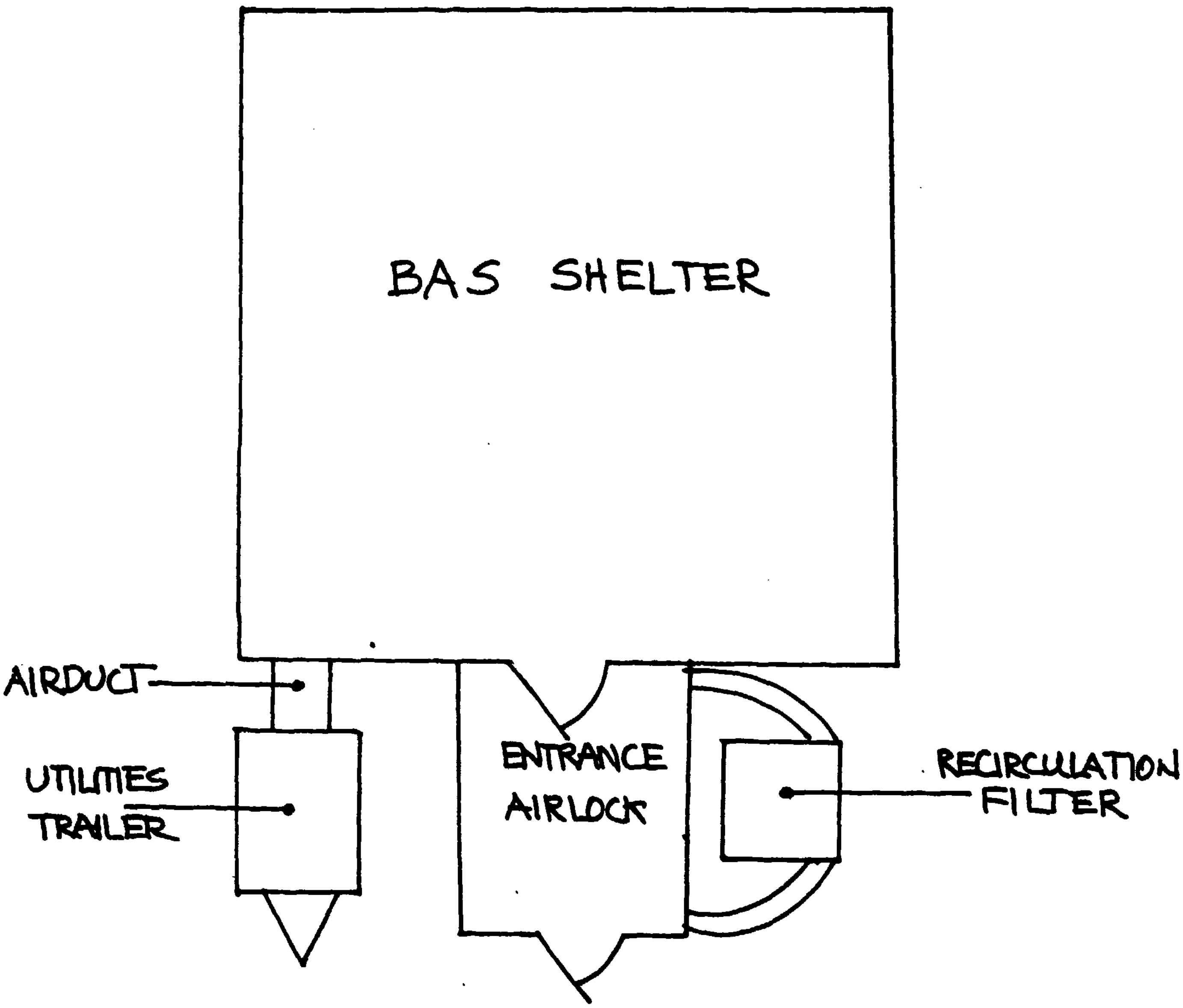


Figure 7. BAS - Proposed Operational Configuration.

Aside from repositioning the trailer, the previous BAS operational configuration remains intact.

3. CONCLUSIONS

As a result of the research and analysis for this effort, several viable conclusions can be drawn:

1) It is possible to provide ample utilities support capability to the new pressurized rib shelter system which incompasses approximately 50% greater square footage than the previous M51 CB POD System.

2) The electrical load, heating, cooling, chemical protection/filtration, and pressurization requirements of the BAS can be attained by replacing some existing equipment with newer, state-of-the-art equipment and by utilizing several proven components from the predecessor utilities package.

3) The proposed utilities package will meet established operational parameters with respect to outdoor/indoor climate (environmental) conditions, chemical protection and power supply.

4) The proposed utility package design can use the M101A1 cargo trailer (modified) to meet allowable vehicle usage requirements with respect to weight limitations and dimensional constraints.

5) The M101A1 cargo trailer bed and bows must be modified to attain the imposed 1500 pound cross-country pay load constraints.

6) The concept of two interconnected BASs forming a Division Clearing Station (DCS) will require two separate utilities packages (one for each BAS), an additional source of electrical power and a power distribution capability.

7) A comprehensive and realistic power loading profile must be accomplished for the proposed DCS concept prior to using any quantified data to determine final equipment and component requirements.

4. RECOMMENDATIONS

Based on the evaluations and analyses completed for each of the subsystems and associated components, the following recommendations are presented for consideration:

4.1 The M101A1 Cargo Trailer

1) Remove the steel bed, including tailgate, wood sides and wood bows, leaving the frame, axle and suspension as it now exists.

2) Fabricate and install an aluminum flatbed on the frame. The flatbed may have raised edges, but not raised enough to interfere with maintenance/operation. The flatbed will be of sufficient strength to support the proposed utilities package equipment.

3) Fabricate aluminum bows to support the canvas in the same manner as presently supported by wood bows.

4.2 The power package. Purchase and install an Isuzu diesel engine, model number QT-23 producing a continuous horsepower rating Of 22.5 at 3600 RPM. It will be water cooled, with a dry weigth of 242 pounds.

4.3 The refrigeration compressor

1) For the BAS design use a Thermo-King compressor, model 0214 with a capacity rating of 41,000 BTUH at 7.75 horsepower at 1500 RPM, and 56,155 BTUH at 11.2 horsepower at 2000 RPM.

2) Use R22 refrigerant.

4.4 The electric generator. Use the same generator and voltage regulator currently adopted for the M51 POD System. This generator is a two bearing, two pole, drip-proof belt driven unit. Ratings include: 208 Volt, 3 Phase, 60 HZ, 5 KW at a power factor of 0.80 continuous duty.

4.5 The mechanical drive design concept

1) A belt drive system should be used to operate the generator and compressor speed reduction mechanism and pulleys. It should be employed as required to attain optimum operating conditions and capacities. Also, a timing belt should be used to drive the generator.

2) Use an electrically operated clutch in the compressor drive system, mounted on the engine shaft. Clutch voltage should be 28 volts.

4) Use pulleys which are large enough to reduce belt tension and are small enough to provide good belt life.

4.6 The environmental control subsystem (ECS)

1) Construct the ECS cabinet with 4mm ALUCOBOND.

2) Increase the diameter of the air delivery ducts from 12 inches to 16 inches.

3) Reduce the length of the air delivery ducts from approximately 16 feet to eight feet, with no bends.

4) Use the Stewart-Warner electrically controlled, multi-fuel heater, model no. 10560M24B1, to meet heating requirements.

5) Use a McQuay Evaporator Coil Type 3-H, finned surface of 20 inches x 21 inches, 4 row, 16 fins per inch with 430 feet per minute face velocity and a capacity of 43,688 BTUH.

6) Use the same condenser fan as that employed presently with the M51 POD system.

7) Condenser coil shall be the McQuay Model TC3X 156 20 inces x 21 1/2 inches, producing 2900 SCFM and 5600 BTUH.

8) Use a hot gas bypass circuit with an external equalizer connected to a side mount distributor to be installed upstream of the hot gas bypass valve. Testing will prove the desirability of including or excluding the circuit for final design.

4.7 The chemical protection subsystem

1) Use the same type gas filters presently incorporated into the M51 POD System. Maximum dimensions of the filters in the new system are 21 inches by 22 inches, thus mandating a dimensional reconfiguration of existing filters.

2) Maintain the entranceway airlock gas and particulate filter system presently used with the M51 POD System.

3) Recirculation fan for the ECS shall be the 12 inch diameter fan presently used with the M51 POD System.

4.8 The pressurized rib inflation blower. The pressurization blower selected for the design shall be the two-stage centrifugal 120 V, Motor No. 096-3470-09 from G&S Electric of Carlisle, PA. A total of two will be required.

4.9 Human Factors Engineering. Implement all human factors considerations, particularly those with respect to sound attenuation.

APPENDIX A

Preliminary Review of Commerical Diesel Engines for Use as a Prime Mover for the Prototype Battalion Aid Station and Division Clearing Station

REPORT ON

PRELIMINARY REVIEW OF COMMERCIAL
DIESEL ENGINES FOR USE AS A PRIME
MOVER FOR THE PROTOTYPE BATTALION AID STATION
AND DIVISION CLEARING STATION

Prepared by Michael McGrath, VSE Corporation

Michael W. McGrath

Contract # DAAK70-81-D-0109

25 September 1985

ENCLOSURE 2

CONTENTS

SUMMARY

A preliminary survey by VSE Corporation of small diesel engines for use with the prototype Pressurized Rib Shelter has surfaced a number of possible engines, the selection of which is contingent on an in-depth analysis of engine requirements for the system and of likely candidates. After specific requirements are identified, market research and analysis of identified candidates will continue until an acceptable engine is found.

ANALYSIS OF ALTERNATIVES

To determine what the specific requirements for an acceptable engine are, a review of the specification requirements used to select the current M51 shelter engine, and of the performance characteristics of the current engine, for comparison, was performed. Particulars are presented in attachment 1.

Having determined the general requirements, a market survey was performed to determine the general availability of an acceptable diesel engine, and to identify what kind of data is immediately available for likely candidates and data that must be researched to make a final selection. Eleven companies were contacted which resulted in reviewing sales literature for 22 different engines. Meetings were held between company representatives and VSE engineers and others are tentatively scheduled pending clarification of system requirements. A review of sales literature indicates that extensive research and a detailed analysis is required to determine acceptability of any given engine.

CONCLUSIONS

Extensive research is required to determine the exact requirements for an acceptable diesel engine, and to identify it. Additional market surveys before exact requirements are known would serve no useful purpose since the surveys will require gathering and analyzing specific data for comparison which is not available through normal sales literature.

ATTACHMENT 1

EXISTING SPECIFICATION REQUIREMENTS FOR M51 ENGINE

REQUIREMENTS

The MIL-STD 20 HP Engine built in accordance with MIL-E-0062014B (ME), 19 August 1968 must be capable of performing in accordance with Par 3.1: "The engines shall be capable of performing as specified herein in any ambient temperature from plus 120°F to minus 25°F; and at any elevation from sea level and a minimum ambient temperature of 120°F, to 5,000 feet and a maximum ambient temperature of 107°F.

Par 3.6.1 STARTING AND OPERATING. The engine shall start within 5 minutes and shall operate after 15 minutes of warm-up under any of the conditions as specified herein.

Par 3.72 MODEL 4A084. The Model 4A084 engine with all accessories shall produce not less than 33.3 maximum brake horsepower (see 6.3.1).

Par 6.3.1 MAXIMUM BRAKE HORSEPOWER. The maximum brake horsepower is that power which the engine will produce at wide-open-throttle at any speed within the operating speed range for periods up to 5 minutes of continuous operation.

Par 3.7.2 The engine shall also produce not less than 29.75 HP for 500 hours at wide-open-throttle at 3550-3650 RPM (corrected to standard conditions (see 6.3.1)).

Par 6.3.2 STANDARD OPERATING CONDITIONS. Standard operating conditions are 29.92 inches of mercury barometric pressure and 60°F air temperature.

Par 3.7.2 '...and not less than 20 continuous brake horsepower (see 6.3.3) for 1500 hours continuous operation at 3550-3650 RPM.'

Par 6.3.3 CONTINUOUS BRAKE HORSEPOWER. Continuous brake horsepower is that power which the engine will produce at any speed within the operating range for periods of 1 hour or more of continuous operation.

The 5 minute time requirement for the maximum horsepower rating suggests that engine temperature is one of the controlling factors for this rating. However, the 29.75 HP/500 hour and 20 HP/1500 hour requirements appear to have a wear related controlling factor. Which one of these ratings is appropriate for the BAS System can only be determined from a mission profile, which is not available at this time. Whether each characteristic of the MIL-STD 20 HP engine need be reproduced by the alternate engine or whether the requirements of the BAS System should dictate the requirements of the alternate engine must be determined. This decision will be made from an engineering analysis.

Low Temperature Starting:

Low temperature starting characteristics of a diesel engine will heavily influence the selection of the alternate engine for this -25°F application. The availability of engine fuels and lubricants may determine the feasibility of using a diesel engine in lieu of a gaoline engine. VSE Corporation provides support services to the PATRIOT Missile System. Work under this contract has included conceptual designs to start diesel engines at low temperatures without the aid of low temperature fuels. This task was initiated because of a contention that low temperature fuels would not be available from NATO sources in Europe.

PERFORMANCE CHARACTERISTICS

Dry weight, without electrics, of this engine is 205 lbs. Theoretical loading of the MIL-STD engine in the M51 BAS System is:

- o Heating Cycle

  A. Generator Load

Evaporator Fan	1.0 HP
Air Lock Fan	0.5 HP
Heater and Controls	0.5 kW
Lighting	0.225 kW

  1. $\dfrac{1.5 \text{ HP}}{1.34 \frac{\text{HP}}{\text{kW}}} + 0.725 \text{ kW} = 1.845 \text{ kW}$

  2. $\dfrac{1.845}{.8 \text{ Generator Efficiency}} = 2.31 \text{ kW}$

  3. $(2.31 \text{ kW}) (1.34 \frac{\text{HP}}{\text{kW}}) = 3.09 \text{ HP}$

  B. Blower Load = 3 HP at 4500 RPM

  C. *Losses = 5% of 3.09 HP + 10% of 3 HP = 0.155 + 0.3 = .455 HP

  D. Total Load = Generator Load + Blower Load + Losses = 3.09 HP + 3 HP + .455 HP = 6.545 HP.

- o Cooling Cycle

  A. Generator Load

Condensor Fan	1.5 HP
Evaporator Fan	1.0 HP
Air Lock Fan	0.5 HP
Lighting	0.225 kW

1. $\frac{3\ HP}{1.34\ \frac{HP}{kW}}$ + .225 kW = 2.465 kW

2. $\frac{2.465}{.8\ \text{Generator Efficiency}}$ = 3.08 kW

3. (3.08 kW) $(1.34\ \frac{HP}{kW})$ = 4.14 HP

B. Blower Load: 3 HP at 4500 RPM

C. Compressor Load: 8 HP at 3000 RPM

D. *Losses

1. Generator 5% of 4.14 HP = 0.207 HP
2. Blower 10% of 3 HP = 0.3 HP
3. Compressor 10% of 8 HP = 0.8 HP

E. Total Load = Generator Load + Blower Load + Compressor Load + Losses = 4.14 HP + 3 HP + 8 HP + 1.307 = 16.447 HP

*Losses = 5% for timing belt and 10% for 3V belt.

The M51 BAS System is required to provide engine fuel for 24 hours of operation. Therefore, evaluation of alternate engines should take into consideration:

- o Wet engine weight with required electrics.
- o Weight of fuel for 24 hours of operation.
- o Weight of fuel tank to contain fuel for 24 hours of operation.

The 20 HP MIL-STD engine producing 16.5 HP consumes 54.8 gallons (333 lbs) of gasoline per 24 hours. This is equivalent to a brake specific fuel consumption (BSFC) of 0.843 lb/brake horsepower hour.

DISCUSSION OF REQUIREMENTS

There are few available specifics relative to altitude requirements and temperature values at altitude. The final report, No. 0415-19, Development of the M51 Collective Protection Shelter System (CB Pressurized POD System) November 1972, Page 145 offers some guidance:

3-78 COMPATIBILITY, HIGH ALTITUDE, AND CHEMICAL AGENT TESTING. Compatibility, high altitude, and chemical agent testing was performed on ED unit ED-1 at Dugway Proving Ground, Utah.

3-80 HIGH ALTITUDE. Following the compatibility testing, the unit was pulled 200 miles Bryan Head Resort near Cedar City, Utah. There at an altitude of 10,400 feet the unit was satisfactorily operated at 3600 RPM. The engine speed was then reduced to 3000 RPM and all systems, including the air conditioning system, performed very well.

This information does not indicate ambient temperature or engine horsepower required at this condition.

3-80 HIGH ALTITUDE. Following the compatibility testing, the unit was pulled 200 miles Bryan Head Resort near Cedar City, Utah. There at an altitude of 10,400 feet the unit was satisfactorily operated at 3600 RPM. The engine speed was then reduced to 3000 RPM and all systems, including the air conditioning system, performed very well.

This information does not indicate ambient temperature or engine horsepower required at this condition.

ATTACHMENT 2

EVALUATION

ENGINES EVALUATED -

Includes all engines for which sales literature were received. Many others are pending receipt of data.

<u>Manufacturer</u>

Number of Cylinders	Model Number	Horse Power	RPM	Dry Weight Engine Only	Method of Cooling
		Lombardini			
2	10LD400-2/B1	15-NA DIN 6270	3600	202	AC
2	9LD560-2	21-NA DIN 6270	3000	242	AC
2	8LD600-2	21-NA DIN 6270	3000	282	AC
2	8LD665-2	24-NA DIN 6270	3000	286	AC
2	8LD740-2	23.8-NA DIN 6270	2600	290	AC
2	5LD675-2	24-NA DIN 6270	3000	451	AC
2	5LD825-2	27-NA DIN 6270	2600	430	AC
		MWM Murphy			
2	D 302-2	Continuous Duty 26 HP	3000	495 430	AC
2	D 202-2	Continuous Duty 26 HP	3000	530	WC
		Farymann Diesel			
2	71A437	18.5-NA DIN 6270	3000	312	AC
2	95A437	22-NA DIN 6270	2500	415	AC
		Petter			
2	P 600-2	Continuous Duty 18.6 HP	3000	397	AC
3	P 600-3	Continuous Duty 38 HP	3000	494	AC
		Isuzu			
2	QT-15	Continuous Power 14.5 HP	2600	209	WC
3	QT-23	Continuous Power 22.5 HP	3600	242	WC
3	QT-35	Continuous Power 31.3 HP	3600	290	WC

ENGINES EVALUATED (Continued)

Number of Cylinders	Model Number	Horse Power	RPM	Dry Weight Engine Only	Method of Cooling
		Wisconsin			
2	W02-1000	Gross Intermittent HP 21.0 Gross x .85 = 17.85	3000	234	AC
		Deutz			
2	F2L511	Continuous Duty 21.8 KW 29.5 HP	3000	340	AC
		Allis Chalmer			
2	Model 213	Continuous Duty 27 HP Generator Rating 29 HP	3000	397	WC
		Lister			
2	LV2	Continuous Duty 18 HP	3600	286	To 125°F AC
2	TS2	Continuous Duty 22 HP	3000	407	To 125°F AC
2	TL2	26.9 HP	3000	429	

NOTE: Alturdyne, a San Diego based company, engaged in power generator design of gas, diesel and turbine power packages, has met with VSE and will provide additional information on request.

DISCUSSION OF EVALUATION

In the past, most U.S. manufacturers expressed engine power in terms of S.A.E. Gross Horsepower. This expression generally was the maximum horsepower a laboratory type engine (an engine minus the accessories not absolutely necessary to the running of the engine) could develop. Power absorbing accessories normally deleted for S.A.E. Gross ratings include air cleaners, mufflers, exhaust systems, and any imposed electrical loads; also in the case of liquid cooled engines, the fan and radiator. S.A.E. Gross test conditions also allow auxilliary cooling of manifolds and negative depression of exhaust collecting systems, both of which will enhance dynamometer test results. All of this points to the fact that S.A.E. Gross Horsepower ratings are not indicative of the power available under the conditions which are normally present for most applications.

The established German D.I.N. standard and the newly adopted S.A.E. Net are similar rating systems in that both rating systems express the horsepower actually available at the fly-wheel when equipped with all accessories required for normal operation. However, differences exist between D.I.N. and S.A.E. Net standards which tend to prohibit direct comparison:

These differences exist because (1) the metric horsepower work unit is only .986 of English horsepower, (2) the atmospheric and temperature base used for corrections are not the same.

NOTE: D.I.N. BASE - 29.92" Hg. at 68°F
S.A.E. NET BASE - 29.35" Hg. at 85°F

There are also different horsepower ratings dependent on various operating conditions and derating factors.

The following excerpts from vendor literature illustrate the degree of analyses required in order to compare horsepower ratings:

<u>Lombardini</u>

DIN RATINGS

- N AUTOMOTIVE RATING: Intermittent duty at variable speed and load. Rating only on request.
- NB RATING NO OVERLOAD CAPACITY: For continuous light duty with constant speed and variable load.
- NA CONTINUOUS RATING OVERLOAD CAPACITY: For continuous heavy duty with constant speed and load. (Ratings certified within 5% after run-in with standard air cleaner and muffler. Derating 1% with standard air cleaner and muffler. Derating 1% approx. every 100 m. altitude and 2% approx. every 5°C above 20°C).
- a Continuous duty. For service beyond application limits, contact Lombardini.

MWM Murphy

Continuous and intermittent engine performance in accordance with S.A.E. Code J270, 85°F (29°C) and 29.38 In. Hg (99.2 kPa). Vehicle performance in accordance with DIN 70020.

Engine output can be demonstrated within 5% at the factor under standard conditions. Deration for temperature is approximately 2.5% for each 10°F (5.8°C) over 85°F (29°C) - for altitude 3% for each 1000 ft. (305m).

Farymann Diesel

Power Data:

DIN 70020 Standard Reference Conditions: Air pressure 760 Torr (sea level), ambient temperature 20°C (68°F).

DIN Ratings:

Ratings certified within 5% after run in with standard air cleaner and muffler. Derating 1% approximately every 100m (330 ft) altitude and 2% approximately every 5°C above 20°C (9°F above 68°F).

Petter

Powers quoted apply to run in engines fitted with air cooling fan, lubricating oil pump, air cleaner and exhaust silencer in accordance with BS 5514/1 or DIN 6271 (ISO 3046/1).

'A' - Continuous Power is equivalent to ISO Standard Power.
'B' - Overload Power is 110% of Continuous Power and available for 1 hour in any 6 hour period of variable load operation, depending on the application.
'C' - Automotive Power, as shown on the graph, relates to variable speed engines and should only be used for transient conditions.

Approximate derating for non-standard site conditions can be obtained by using the following correction factors:

Altitude 6 1/2% per 500m above 150m. Temperature 3% per 10°C above 27°C. For accurate values of derating consult Petters Limited.

Idna North American Isuzu Diesel

Notes accompanying engine performance curves:

1. Performance is with alternator unloaded, without fan, without intake or exhaust restriction and with No. 2 diesel fuel .853 SG at 60°F but derated to 100°F for SAE J816B and 104°F for SAE J1349.

2. Refer to EX3-0.0-000-1031 for exhaust and air intake restriction corrections and to FN0.0-300-1158 and FN0.0-000-1010 for fan parasitic losses to obtain net ratings.

Notes accompanying tabulated data:

Rated intermittant power according to (JIS D 1005) continuous power according to (JIS 88013) (JIS 88014)

Wisconsin

Engine Rating Conditions: Engines tested per SAE J1349 are gross intermittent power ratings; engine equipped with cooling fan, muffler, and air cleaner, corrected to standard conditions of 29.31 in. Hg (99 kPa) dry barometer and 77°F (25°C) temperature. Engine outputs can be demonstrated within 5% at factory under rating conditions. Values are for standard engines.

Deutz

Specification Data:

1. Continuous output "A" (10% overload) and intermittent output "B" heavy duty, to DIN 6270.

2. Automotive output to DIN 70020 and intermittent output "B" light duty to DIN 6270.

Allis-Chalmer

Allis-Chalmers diesel engines are rated at 85°F and 29.38" Hg (500 ft. altitude). There is no horsepower loss at rated speed for turbocharged or intercooled engines up to 5,000 feet (1.524m) and in some instances to 10,000 feet (3,048m). Fuel reduction required on some models above 7,500 feet (2,286m). On naturally aspirated engines, horsepower loss occurs at 1,000 feet (305 m) at standby rating and 1,500 feet (457 m) at prime rating.

Curve 1 Represents the power available at full throttle for applications in which the engine will operate under highly variable conditions of load and speed. Factory approval required.

Curve 2 Recommended power for variable load applications where full throttle operation might be required for extended periods ... followed by equal periods of operation at reduced loads.

Curve 3 Recommended power to be used for driving sustained loads for continuous-duty operation.

Standby Power Rating - the power output at which an engine may operate for the duration of a commercial power outage.

Prime Power Rating - the power output at which an engine normally operates, with an overload capability for operating at a power output of up to the standby rating for intermittent periods.

Lister

A = continuous bhp
B = Intermittent power to DIN 6270 "B"
C = Maximum gross bhp

These figures apply to fully run in non-derated bare engines without power absorbing extras, transmissions, gear boxes etc., built, set and tested for each of the speeds shown.

Rating:

Note that 10% overload and DIN B ratings apply only to a fully run in engine. This is normally attained after a period of approximately 50 hours running, but if specifically negotiated, engines can be supplied delivering these outputs Ex works.

Derating:

Altitude - 3 1/2% for every 1000 above 500 ft. above sea level.

Air inlet temperature - 2% for every 10°F above 85°F.

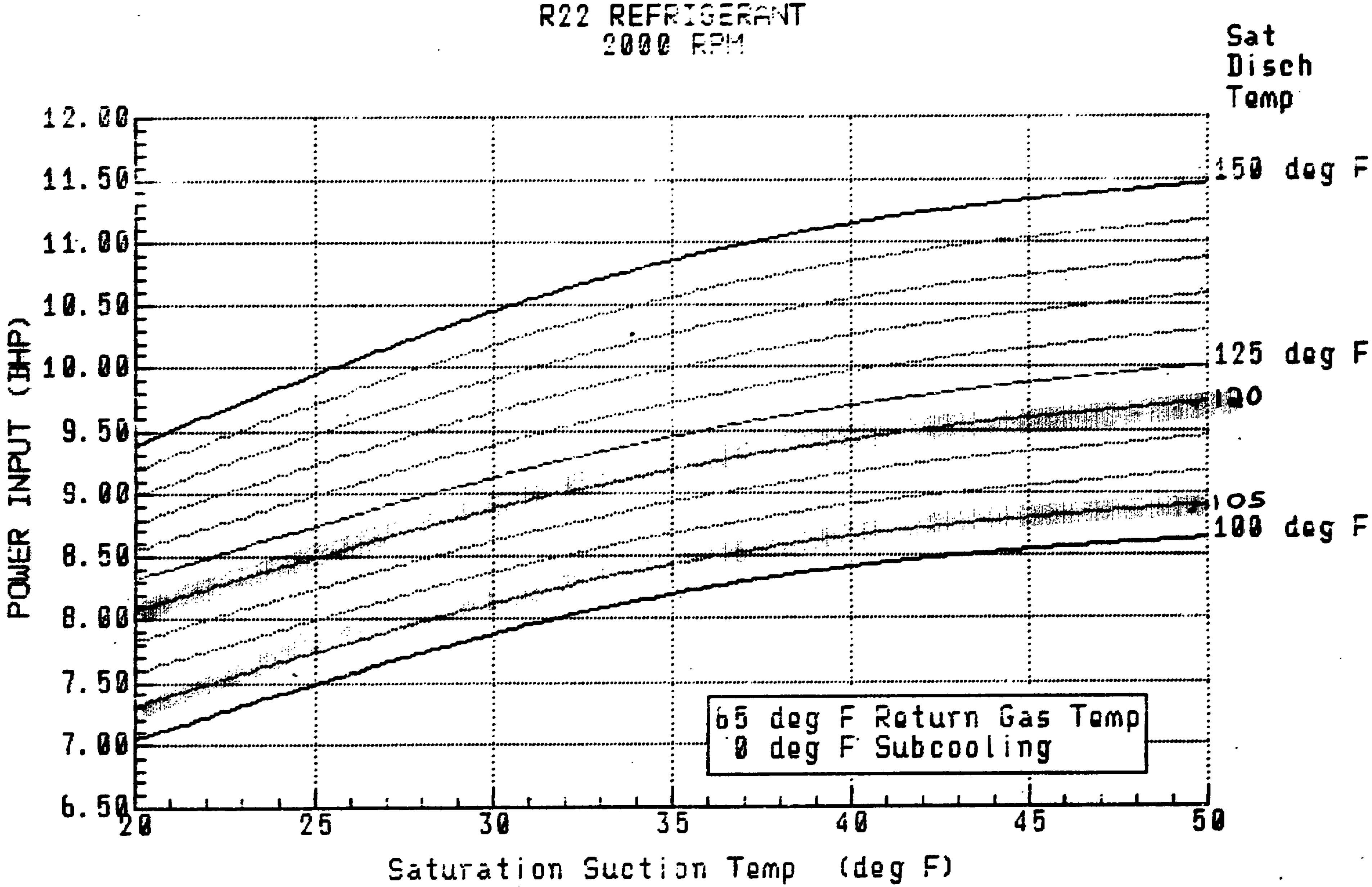
Humidity:

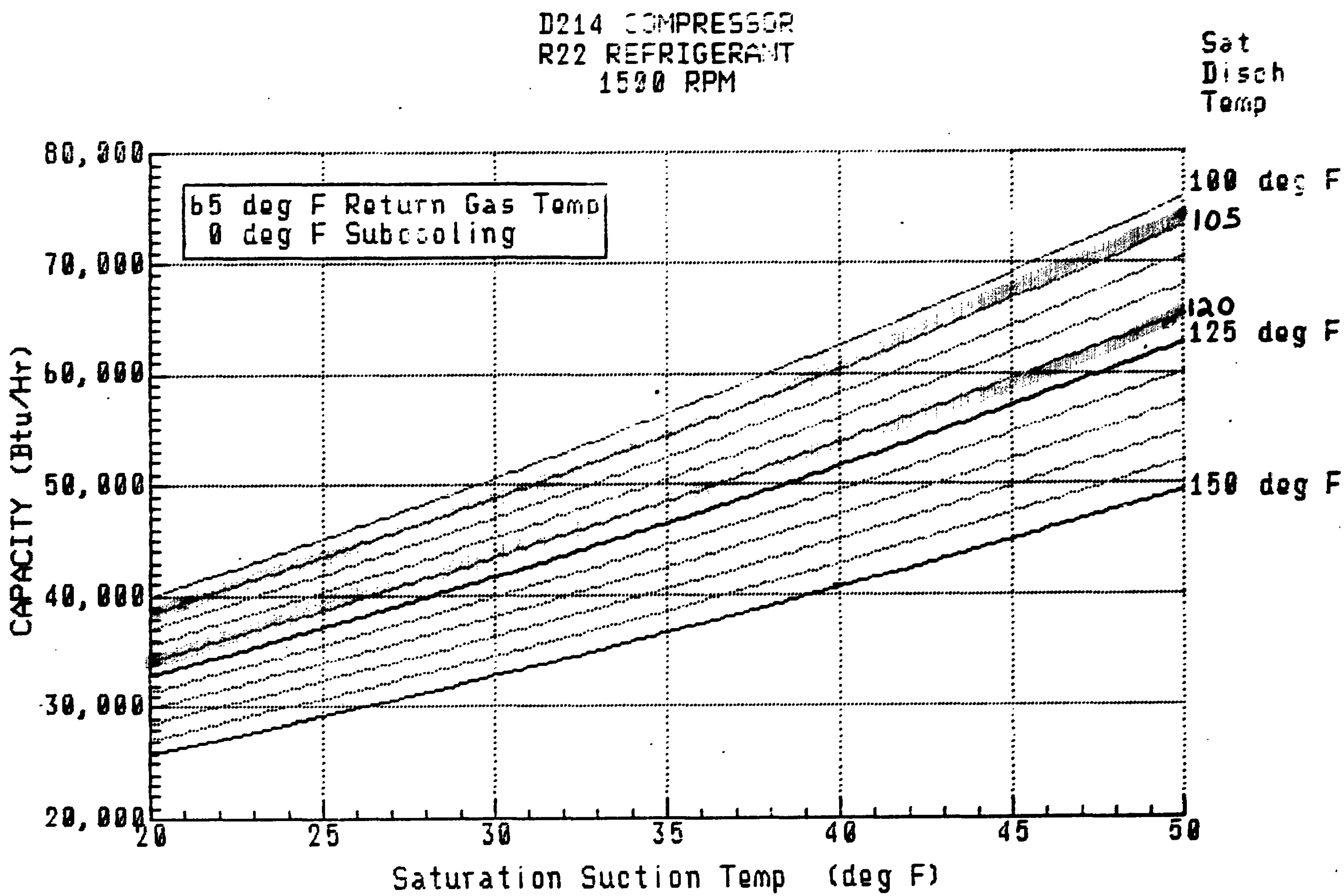
Up to a maximum of 6%.

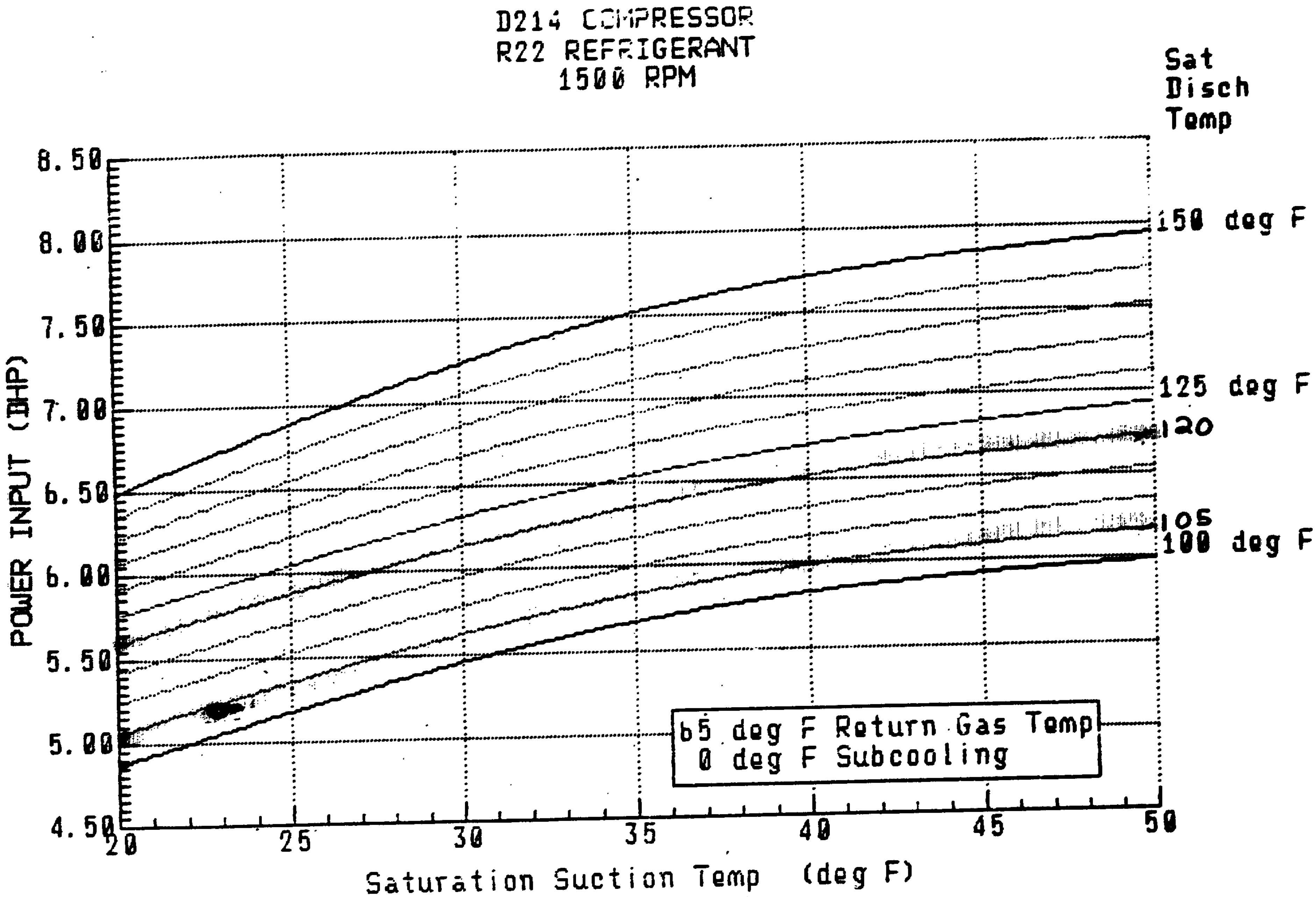
APPENDIX B

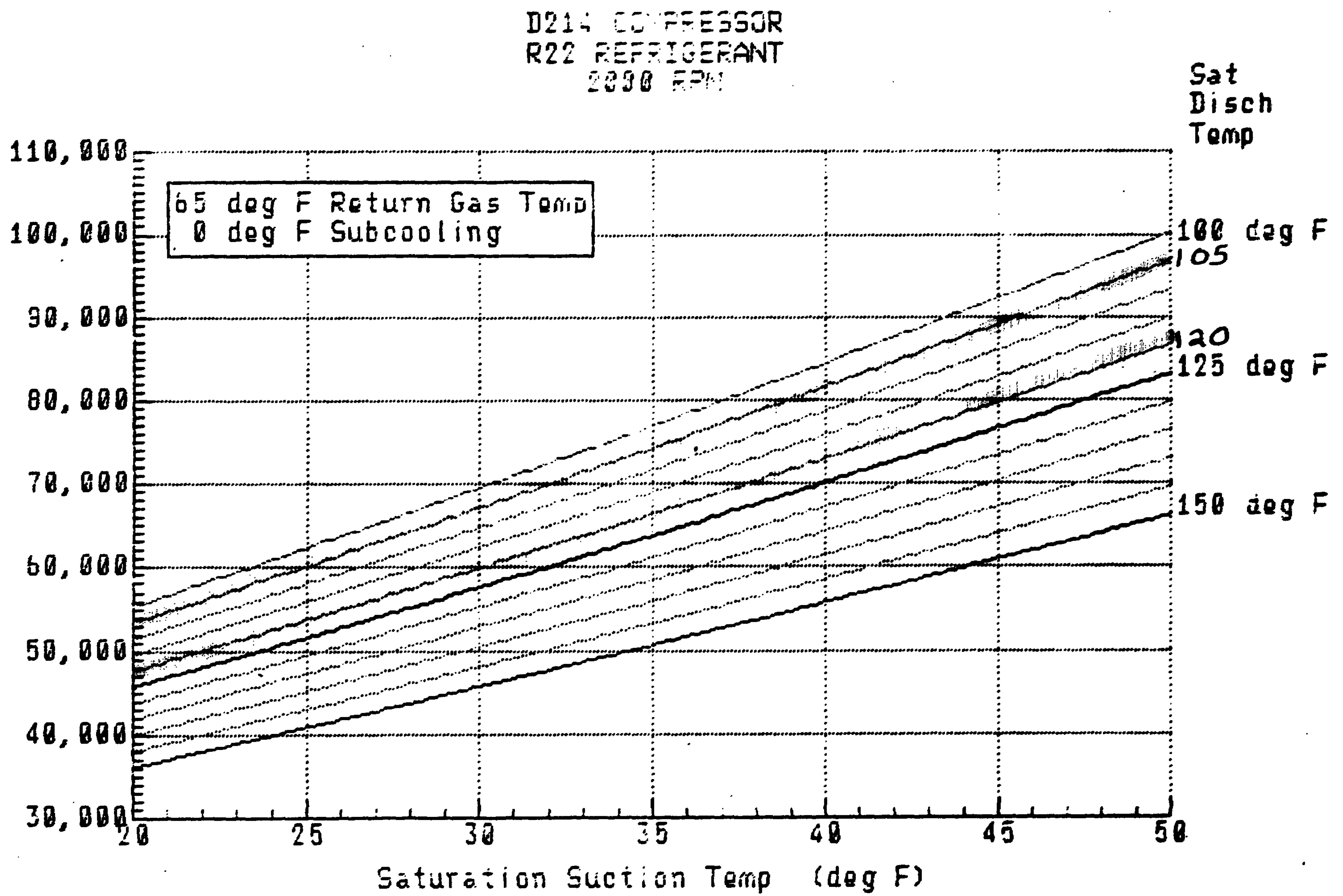
Capacity and Horsepower Curves for Thermo-King Model D-214 Compressor

D214 COMPRESSOR
R22 REFRIGERANT
2000 RPM




D214 COMPRESSOR
R22 REFRIGERANT
1500 RPM
Sat
Disch
Temp
65 deg F Return Gas Temp
0 deg F Subcooling
100 deg F
105
120
125 deg F
150 deg F
80,000
70,000
60,000
50,000
40,000
30,000
20,000
CAPACITY (Btu/Hr)
20
25
30
35
40
45
50
Saturation Suction Temp (deg F)


D214 COMPRESSOR
R22 REFRIGERANT
1500 RPM
Sat
Disch
Temp
150 deg F
125 deg F
120
105
100 deg F
8.50
8.00
7.50
7.00
6.50
6.00
5.50
5.00
4.50
POWER INPUT (BHP)
20
25
30
35
40
45
50
Saturation Suction Temp (deg F)
65 deg F Return Gas Temp
0 deg F Subcooling



D214 COMPRESSOR
R22 REFRIGERANT
2000 RPM
65 deg F Return Gas Temp
0 deg F Subcooling
Sat
Disch
Temp
100 deg F
105
120
125 deg F
150 deg F
CAPACITY (Btu/Hr)
110,000
100,000
90,000
80,000
70,000
60,000
50,000
40,000
30,000
20
25
30
35
40
45
50
Saturation Suction Temp (deg F)

APPENDIX C

Cooling Load Calculations

NOTE: These calculations have been extracted from the Final Report of "Analysis of the Heating and Cooling Loads for the M51 CB POD System and the Pressurized Rib Shelter" and modified to reflect updated information.

33

CONDUCTANCE FORMULAS AND FACTORS FOR THE PRESSURIZED RIB SHELTER

The Pressurized Rib (PR) shelter incorporates an insulated liner within the air pressurized rib supported outer skin. In addition to the same insulating surface films and air spaces of the M51 shelter, a 2" fabric lined fiberglass wool insulating batt is hung from the shelter walls and roof and is used as an insulating liner. (See Figure 4 and 12)

Total conductance is equal to:

$$U_T = \frac{1}{\frac{1}{h_i} + \frac{1}{C_i} + \frac{1}{C_a} + \frac{1}{h_o}} \qquad \text{Eq 27}$$

where h_i, C_a and h_o are the same symbols and values as used in the M51 shelter and

C_i = conductance value for the 2" insulating batt proposed by the U.S. Army Natick R&D Center (NRDC)

6.0

The NRDC given value for this insulating batt is R = ~~5~~.0

$$C_i = \frac{1}{R} = \frac{1}{\cancel{5}} = 0.20$$

$$6 = 0.17$$

C-1

The wall/roof construction of the Pressurized Rib shelter is thermally equivalent to that of the M51-C8 Pod with the exception of the added insulating bat. Otherwise, the materials are thermally the same, the surface films are the same in number and conductance values, and the conductivity of the air spaces can be considered the same. (Note that after an air space exceeds a certain width, depending on orientation and flow direction, convection currents tend to prevent a further reduction of the conductance values.)

Therefore, we can modify the U_1 numbers from the M51 walls as follows:

The average summer U_1 for the M51 shelter arch = 0.59 (from page B-16)

The Pressurized Rib arch summer $U_{1a} = \dfrac{1}{\dfrac{1}{0.59} + \dfrac{1}{C_1}}$

$$= \frac{1}{\dfrac{1}{0.59} + \dfrac{1}{\cancel{0.20}\ 0.17}} = \cancel{0.15}\ 0.13 \text{ Btu/(hr x sq ft x °F)}$$

The summer U_1 for the M51 shelter ends = 0.65 (from page B-16)

The Press. Rib end summer U_{1s} =

$$\frac{1}{\dfrac{1}{0.65} + \dfrac{1}{\cancel{0.20}\ 0.17}} = \cancel{0.15}\ 0.13 \text{ Btu/(hr x sq ft x °F)}$$

For winter U_i values of the Pressurized Rib (PR) Shelter, modify the M51 U_i values on page B-37 as follows:

$$\text{PR arch } U_{iw} = \frac{1}{\frac{1}{0.52} + \frac{1}{\cancel{\text{[illegible]}}\ 0.17}} = 0.14\ \text{Btu/(hr x sq ft x °F)}\quad 0.13$$

$$\text{PR end } U_{iw} = \frac{1}{\frac{1}{0.49} + \frac{1}{\cancel{0.20}\ 0.17}} = 0.14\ \text{Btu/(hr x sq ft x °F)}\quad 0.13$$

For summer heat balances to determine the outer skin material temperature for the PR Shelter, the following data will be needed:

I_{DN} = 313 for the 120°/80° condition	(see page B-7)
I_{DN} = 302 for the 105°/80° condition	(see page B-7)
Angle E = 57.3° for the PR arch	(see page B-4)
Angle E = 90° - 14° = 76° for the PR Arch	(see page B-4)
h_c = 3.11 (summer)	(from page B-17)
t_o = 120°F (summer hot-dry)	(Ref Appendix A)
t_o = 105°F (summer hot-humid)	(Ref Appendix A)
t_i = 80°F (summer)	(Ref Appendix A)
T_g = 605°R (summer hot-dry)	(Ref Appendix A)
T_g = 590°R (summer hot-humid)	(Ref Appendix A)
T_a = 580°R (summer hot-dry)	(from page B-12)
T_a = 565°R (summer hot-humid)	(from page B-12)
Arch A = 670 sq ft.	(from page B-4)
End A = 344 sq ft.	(from page B-5)
Floor A = 462 sq ft.	(from page B-4)

Summer Heat Balance on PR Arch at 120°/80° (Wet-Dry)

Using equation 22 (page B-18).

$313(0.282 + 0.513 \cos 57.3°)$

$= 2.458 \times 10^{-10} (T_m^4 - 605^4)(1 - \cos 57.3°)$

$+ 5.996 \times 10^{-10} (T_m^4 - 580^4)(1 + \cos 57.3°)$

$+ 3.11 (t_m - 120°)$

$+ \cancel{0.15}\ 0.13 (t_m - 80°)$

try $t_m = \cancel{161.2°F}$ 161.5°

3 129.2 10.8

$175.0 = 1.7 + 33.\cancel{8} + 128.1 + \cancel{12.2}$

$175.0 = 175.0$

then $t_m = 161.\cancel{2}5°F$ for the PR arch at 120°/80° and the arch heat load

$= \cancel{12.2}\ 10.8 \times 670$

$= \cancel{8174}$ Btu/hr

7236

Summer Heat Balance on PR Ends for 120°/80° Condition

$313(0.282 + 0.513 \cos 76°) = 2.458 \times 10^{-10} (T_m^4 - 605^4)(1 - \cos 76°)$

$+ 5.996 \times 10^{-10} (T_m^4 - 580^4)(1 + \cos 76°)$

$+ 3.11 (t_m - 120°)$

$+ \cancel{0.16}\ 0.13 (t_m - 80°)$

try $t_m = \cancel{160.9°F}$ 151.2°

7 7 9.4

$127.1 = 1.0 + 19.\cancel{8} + 9\cancel{6}.0 + \cancel{10.6}$

$127.1 = 127.0$

then $t_m = \cancel{160.8}\ 151.8°F$ for the PR end at 120°/80° and the end heat load

$= \cancel{10.6} \times 344 = \cancel{3646}$ Btu/hr

9.4 = 3234

Heat Gain through the PR Shelter Floor

The floor will be treated in two ways. First the heat load for an uninsulated floor in contact with the ground and, second, with ~~a 1"~~ A ~~polyurethane board~~ floor with a conductance value OF R4 TO R5. WE WILL CALCUL-ATE FOR A FLOOR WITH A CONDUCTANCE VALUE OF R4.5

$C_i = \frac{1}{R} = \frac{1}{\cancel{6.25}\ 4.5} = 0.16$ Btu/(hr x sq ft x °F) $= .22$ Ref 1P

In the first case the heat transfer factor for the floor $= h_i = 1.63$ (from page B-21)

In the second case the heat transfer factor for the floor

$$U_f = \frac{1}{\frac{1}{1.63} + \frac{1}{\cancel{0.16}\ 0.22}} = \cancel{0.16} \text{ Btu/(hr x sq ft x °F)} = 0.19$$

Floor perimeter area = $2 \times 2 \frac{(252 + 264)}{12} = 172$ sq ft

Floor heat gain (No insulation)

$q_{fs} \approx h_i \times A \times (t_o - t_i)$ (Eq 238, page B-22)

$= 1.63 \times 172\ (120° - 80°)$

$= 11,214$ Btu/hr

Floor heat gain (~~with 1" insulation~~) WITH A FLOOR WHICH HAS A CONDUCTANCE VALUE OF 0.19

$q_{fs} \approx\ _i \times A \times (t_o - t_i)$

$= \cancel{0.15}\ 0.19 \times 172\ (120° - 80°)$

$= \cancel{1032}$ Btu/hr

1307 BTU/HR

Duct Heat Gain at 120°/80°

Same as on page B-29 except with 47.1 sq ft area

Heat gain = $0.42 \times 47.1\ (145° - 70°)$

$q_d = 1484$ Btu/hr

Ventilation Air Heat Load at 120°/80°

Same as on page B-30

Sensible q_{as} = 5875 Btu/hr

Latent Q_{al} = -6775 Btu/hr = 0

Evaporator Compartment Heat Load at 120°/80°

Same as on page B-30 except with 20 sq ft of surface

q_{es} = 0.50 x 20 (145° - 70°)

= 750 Btu/hr

Evaporator Fan Heat Load

Same as on page B-25

q_f = 3183 Btu/hr

VETILATION FAN
~~Pressure Blower~~
400 WATTS

~~Same as on page B-25~~

q_b = WATTS x 3.41

q_b = ~~3463~~ Btu/hr
1364

Lights

Heat gain, q_l = 1705 Btu/hr

People at 120°/80°

Same as on page B-41

Sensible, q_{ps} = 2200 Btu/hr

Latent, q_{pl} = 1800 Btu/hr

Summary - Pressurized Rib Shelter

Heat Gain at 120°/80°

Sensible Load:

Shelter (~~No~~ WITH floor insulation)	~~23,034~~ 11,777 Btu/hr
Ducts	1,484
Ventilation Air	5,875
Evaporator Compartment	750
Evaporator Fan	3,183
~~Pressure Blower~~ VENTILATION FAN	~~3,463~~ 1364
Lights	1,705
People	2,200
Total Sensible =	~~41,694~~ Btu/hr 28,338

Latent Load:

Ventilation Air	-6,775
People	1,800
Total Latent =	0 Btu/hr

Total Cooling Requirement	~~41,694~~ Btu/hr 28,338

Summer Heat Balance on PR Arch at 105°/80°

Using equation 22 (page B-18) and data from page B-45:

$$302(0.282 + 0.513 \cos 57.3°)$$

$$= 2.458 \times 10^{-10} (T_m^4 - 590^4)(1 - \cos 57.3°)$$

$$+ 5.996 \times 10^{-10} (T_m^4 - 565^4)(1 + \cos 57.3°)$$

$$+ 3.11 (t_m - 105°)$$

$$+ \cancel{0.16}\ 0.13 (t_m - 80°)$$

try $t_m = \cancel{145.9°F}\ 146.2°$

$$168.9 = 1.6 + 30.6 + \cancel{127.0}\ 128.0 + \cancel{9.9}\ 8.7$$

$$168.9 = \cancel{168.8}\ 168.9$$

Then t_m = 146.2°F for the PR shelter arch at 105°/80° and the arch heat load =

~~9.9~~ 8.7 x 670 = 6633 Btu/hr = 5829

Summer Heat Balance on PR Ends at 105°/80°

$$302(0.282 + 0.513 \cos 76°)$$

$$= 2.458 \times 10^{-10} (T_m^4 - 590^4)(1 - \cos 76°)$$

$$+ 5.996 \times 10^{-10} (T_m^4 - 565^4)(1 + \cos 76°)$$

$$+ 3.11 (t_m - 105°)$$

$$+ \cancel{0.16}\ 0.13 (t_m - 80°)$$

try $t_m = \cancel{136.7°F}\ \cancel{135.96°}\ 136.0$

$$122.6 = 0.9 + \cancel{17.8}\ 18.0 + \cancel{95.6}\ 96.3 + \cancel{8.4}\ 7.4$$

$$122.6 = \cancel{122.8}\ 122.6$$

Then $t_m = \cancel{136.7°F}$ 136.0 for the PR shelter ends at 105°/80° and the end heat load

= 7.4 x 344 = 2546 Btu/hr

Heat Gain through the PR Shelter Floor at 105°/80°

Floor heat gain (no insulation)

$$q_{fs} = 1.63 \times 172 (105° - 80°) = 7009 \text{ Btu/hr}$$

Floor heat gain (with insulation

$q_{fs} = \cancel{0.15}\ 0.19 \times 172\ (105° - 80°) = \cancel{645}\ 817$ Btu/hr

Duct Heat Gain at 105°/80°

Same as on page B-34 except with 47.1 sq ft area

$q_d = 0.42 \times 47.1\ (133° - 70°)$

$= 1246$ Btu/hr

Ventilation Air Heat Load at 105°/80°

Same as on page B-34

Sensible $q_{as} = 3634$ Btu/hr

Latent $q_{al} = 9833$ Btu/hr

Evaporator Compartment Heat Load at 105°/80°

Same as on page B-35 except with 20 sq ft area.

$q = 0.5 \times 20\ (135° - 70°) = 650$ Btu/hr

Evaporator Fan Heat Load

Same as on page B-25

$q = 3183$ Btu/hr

VENTILATION FAN

~~Pressure Blower Heat Gain~~

Same as on page B-~~34~~ 48

q = ~~8468~~ 1364 Btu/hr

Lights

Same as on page B-26

$q = 1705$ Btu/hr

People at 105°/80°

Same as on page B-31

Sensible = 2200 Btu/hr

Latent = 1800 Btu/hr

4/2

Summary - Pressurized Rib Shelter

Heat Gain at 105°/80°

Sensible Load:

Shelter (~~No~~ WITH floor insulation)	~~16,532~~ 9,192
Ducts	1,246
Ventilation Air	3,634
Evaporator Compartment	650
Evaporator Fan	3,183
~~Pressure Blower~~ VENTILATION FAN	~~3,463~~ 1,364
Lights	1,705
People	2,200
Total Sensible =	~~32,613~~ 23,174 Btu/hr

Latent Load:

Ventilation Air	9,833
People	1,800
Total Latent =	11,633 Btu/hr

Total Cooling Requirement	~~44,246~~ 34,807 Btu/hr

APPENDIX D

Heating Load Calculations

Winter Heat Loss - Pressurized Rib Shelter

The winter inner wall conductances for the M51 shelter are:

U_{ia} = 0.52 (for the M51 Arch) and

U_{ie} = 0.49 (for the M51 ends) (from page B-37)

If we modify these parameters to include the 2" insulating batt as described on page B-45, where $C_i = \cancel{0.20}\ 0.17$ we get new winter U_i values for the PR Shelter as follows:

$$U_{ia} = \frac{1}{\frac{1}{0.52} + \frac{1}{\cancel{0.20}\ 0.17}} = \cancel{0.14}\ 0.13 \text{ Btu/(hr x sq ft x °F) for the arch.}$$

$$U_{ie} = \frac{1}{\frac{1}{0.49} + \frac{1}{\cancel{0.20}\ 0.17}} = \cancel{0.14}\ 0.13 \text{ Btu/(hr x sq ft x °F) for the ends.}$$

h_c = 5.11 (from page B-17)

For the uninsulated floor:

h_i = 1.08 (from page B-21)

For an insulated floor using the same insulation indicated on page B-47:

$$U_{fw} = \frac{1}{\frac{1}{h_i} + \frac{1}{C_i}} = \frac{1}{\frac{1}{1.08} + \frac{1}{\cancel{0.16}\ 0.22}} = \cancel{0.14} = 0.183 \text{ Btu/(hr x sq ft x °F)}$$

115

Winter Heat Balance - PR Shelter Arch

Using equation 26 (page B-38) with values from page B-41, we have:

$$
\begin{aligned}
0 = {} & 2.458 \times 10^{-10} (T_m^4 - 425^4)(1 - \cos 57.3°) \\
& + 5.996 \times 10^{-10} (T_m^4 - 410^4)(1 + \cos 57.3°) \\
& + 5.11 (t_m - (-25°)) \\
& + \overset{0.13}{\cancel{0.14}} (t_m - 70°)
\end{aligned}
$$

In order to balance this equation, try $t_m = -24°$F. $-24.1°$

$$0 = 0.4 + 7.3 + \overset{4.6}{\cancel{5.5}} - \overset{12.2}{\cancel{13.2}}$$

$$0 = \cancel{0} \; 0.1$$

$-23.9°$ $(76°)$

Then the outer surface of the PR shelter arch = -24°F and the heat loss from the arch = $\overset{12.2}{\cancel{13.2}}$ x 670 = 8844 Btu/hr

8174

Winter Heat Balance - PR Shelter Ends

All values in equation 26 are the same as for the arch, except angle E which becomes 76°.

In order to balance this equation, try t_m = ~~-23.7~~°F -23.9

$$0 = 0.7 + 5.9 + \overset{5.7}{\cancel{6.5}} - \overset{12.2}{\cancel{10.4}}$$

$$0 = 0.1$$

Then the outer surface of the PR shelter ends = ~~-23.7~~°F and the heat loss -23.9 from the ends = ~~13.1~~ x 344 = 4506 Btu/hr

12.2 4197

Heat Loss through the Floor - No Insulation

Using the perimeter method from page B-39 but with A = 172 sq ft:

$$
\begin{aligned}
q_{fw} &= 1.08 \times 172 (70° - (-25°)) \\
&= 17{,}647 \text{ Btu/hr}
\end{aligned}
$$

Heat Loss through the Floor - With Insulation

$$q_{fw} = \overset{0.183}{\cancel{0.14}} \times 172\ (70° - (-25°))$$

$$= 2288 \text{ Btu/hr}$$

2710

Duct Heat Loss - PR System:

Same as on page B-39 but with estimated average duct temperature = 80°F:

$$q_{dw} = 0.44 \times 47.1\ (80° - (-25°)) = 2176 \text{ Btu/hr}$$

Ventilation Air Heat Loss

Using equation 24 (page B-24) as on page B-42, q_{aw} = 13,673 Btu/hr

Evaporator Compartment Heat Loss:

Same as on page B-40 but with average evaporator air temperature of 80°F:

A = 20 sq ft

$$q_{ew} = 0.50 \times 20\ (80° - 5°)$$

$$= 750 \text{ Btu/hr}$$

<u>Summary - Pressurized Rib Shelter Winter Heat Loss</u>

Arch	8,174 ~~8,844~~ Btu/hr
Ends	4 197 ~~4,506~~
Floor (with ~~No~~ insulation)	2990 ~~17,647~~
Sub Total =	30,997
Ducts	2,176
Ventilation Air	18,673
Evaporator Compartment	750
Total Heat Loss =	36,940 ~~52,596~~ Btu/hr
	with (No floor insulation)

APPENDIX E

Report on Development of the M51 Collective Protection Shelter System

Report No. 0415-19

DEVELOPMENT OF THE M51 COLLECTIVE PROTECTION SHELTER SYSTEM

(CB PRESSURIZED POD SYSTEM)

Final Comprehensive Report

by

David L. Reynolds

NOVEMBER 1972

DEPARTMENT OF THE ARMY
EDGEWOOD ARSENAL
Development and Engineering Directorate
Physical Protection Branch
Edgewood Arsenal, Maryland 21010

DAAA15-67-C-0415

Technical Support Group
DEFENSE PRODUCTS GROUP
AMERICAN AIR FILTER COMPANY, INC.
St. Louis, Missouri 63132

elements. Analysis of the individual filters is contained in the following paragraphs.

2-91. GAS FILTER DESIGN ANALYSIS. In addition to the GPFU filters, two recirculation carbon filters were required for the M51 shelter system; one for the airlock recirculation unit and one for the ECU. Each filter differs from the other in various aspects, the requirements of which are as follows:

a. The airlock recirculation gas filter must conform to the following requirements:

Rated Airflow	550 standard cfm (scfm)
Maximum Pressure Drop	0.6 in. wg
Dimensions.	3 by 16 by 26 in.
Basic Construction Material	Aluminum
Filter Material	12 lbs ASC Whetlerite charcoal conforming to specification MIL-C-13724A

b. The ECU recirculation gas filter must conform to the following requirements:

Rated Airflow	1200 scfm
Maximum Pressure Drop	0.6 in. wg
Dimensions.	3 by 16 by 26 in.
Basic Construction Material	Aluminum
Filter Material	12 lbs ASC Whetlerite charcoal conforming to specification MIL-C-13724A, except that the mesh size shall be 6 by 16 in lieu of 12 by 32 as specified.

2-92. Calculations For Active Face Area. The active face area is that area of the filter which actually experiences airflow; flanges or baffle edges are not included. As noted in paragraph 2-91, the outside filter dimensions shall be 3 by 16 by 26 inches. Subtracting the width of the flanges and multiplying the face dimensions will yield the active filter face area.

$$(26 - 0.75)\ (16 - 0.75) = (25.25)\ (15.25) = 385.1 \text{ in.}^2 \text{ (Active face area)}$$

$$\text{or, } \frac{385.1}{144} = 2.67 \text{ ft}^2 \text{ (Active face area)}$$

E-2

2-93. Calculations for Active Tray Area. Active tray area may be found by means of a similar calculation. Nominal tray dimensions are 2-7/8 by 25-3/4 inches; since active tray area is also defined as that area of the tray that experiences airflow, subtracting the baffle edge around the perforations yields the active tray dimensions. The area is determined by multiplying the results.

Active tray length = 25.25 in.

Active tray width = 2.75 in.

$$\text{Active tray area} = \frac{2.75 \times 25.25}{144} = 0.482 \text{ ft}^2$$

2-94. The number of trays in a filter is 38. This number was arrived at by empirical methods and was conducive to low pressure drop. The total active tray area was therefore calculated:

$$\text{Total active tray area} = 38 \times 0.482 = 18.3 \text{ ft}^2$$

2-95. Calculations For Bed Depth And Tray Thickness. The following calculations indicate the bed depth necessary to accommodate 12 pounds of charcoal in the given trays.

Length of tray occupied by charcoal = 25.75 in.

Width of tray occupied by charcoal = 2.875 in.

Area of tray occupied by charcoal = 25.75 x 2.875 = 74.03 in.^{2}

Total tray area occupied by charcoal = 38 x 74.03 = 2813 in.^{2}

Carbon required = 12 lbs (design specification)

$$\text{Volume of carbon} - \frac{12 \text{ lbs}}{0.02 \text{ lbs/in.}^3} = 600 \text{ in.}^3$$

$$\text{Carbon bed depth} = \frac{600 \text{ in.}^3}{2813 \text{ in.}^2} = 0.213 \text{ in.}$$

2-96. Pressure Drop Calculations For 12 By 32 Mesh Charcoal. The following formula has been developed to determine pressure drop through the charcoal filter.

$$P = 8.53 \times 10^{-9} \left(\frac{1}{m} \quad (1 + \frac{Z}{h})\right) Vf^2 + 0.0289\, Z \left((1 + \frac{Z}{h} \quad \frac{m}{4(1+m)^2} + \frac{m}{(1+m)}\right) V_f$$

Where P = Pressure drop, inches wg.

m = h/2L

L = Length of channel, inches

Z = Outside tray thickness, inches

1 = Channel height, measured between trays at the center of the V-pleat, inches

V_f = Velocity at filter face, fpm

Note. This equation could be used for practically any pleated bed; however, the coefficients for each bed must be known. The coefficients 8.53×10^{-9} and 0.0289 have been developed for 12 by 32 mesh charcoal having a fines cloth retainer (minute particle retainer).

2-97. The data for the equation given in paragraph 2-96 are:

$$2L = 2.75 \text{ in.}$$

$$Z = 0.25 \text{ in.}$$

$$h = \frac{15.25 - 38\ (0.25)}{38} = 0.151 \text{ in.}$$

Substituting, $\frac{Z}{h} = \frac{0.25}{0.151} = 1.655 \qquad m = \frac{0.151}{2.75} = 0.0549$

$$P = 8.53 \times 10^{-9} \left(\frac{1}{0.0549} \times (2.655) \right) V_f^2 +$$

$$0.0289\ (0.25)\ (2.655) \left(\frac{0.0549}{4(1.0549)^2} + \frac{0.0549}{(1.0549)} \right) V_f$$

Therefore, $P = 4.125 \times 10^{-7}\ V_f^2 + 1.23 \times 10^{-3}\ V_f$

for 12 by 32 mesh charcoal with fines cloth.

The following values are computed from this equation:

CFM (ft^3/min)	FACE VELOCITY (Q/A, ft/min)	PRESSURE DROP (in. wg)
1200	449	0.64
550	206	0.27

These data are plotted in Figure 2-22.

2-98. Pressure Drop For 6 By 16 Mesh Charcoal. Data were needed on the filter unit having the 6 by 16 mesh charcoal; a test filter was therefore built to study resistance characteristics. This filter was similar in design to the 12 by 32 mesh charcoal filter. The resistance traverse which was run is plotted in figure 2-22 together with the curve for the 12 by 32 mesh charcoal. The two plots are not parallel, and this is because of range inaccuracies in the equation. Nevertheless, insofar as pressure drop is concerned, the plots indicate that the filter design will be good at 550 cfm

E-4

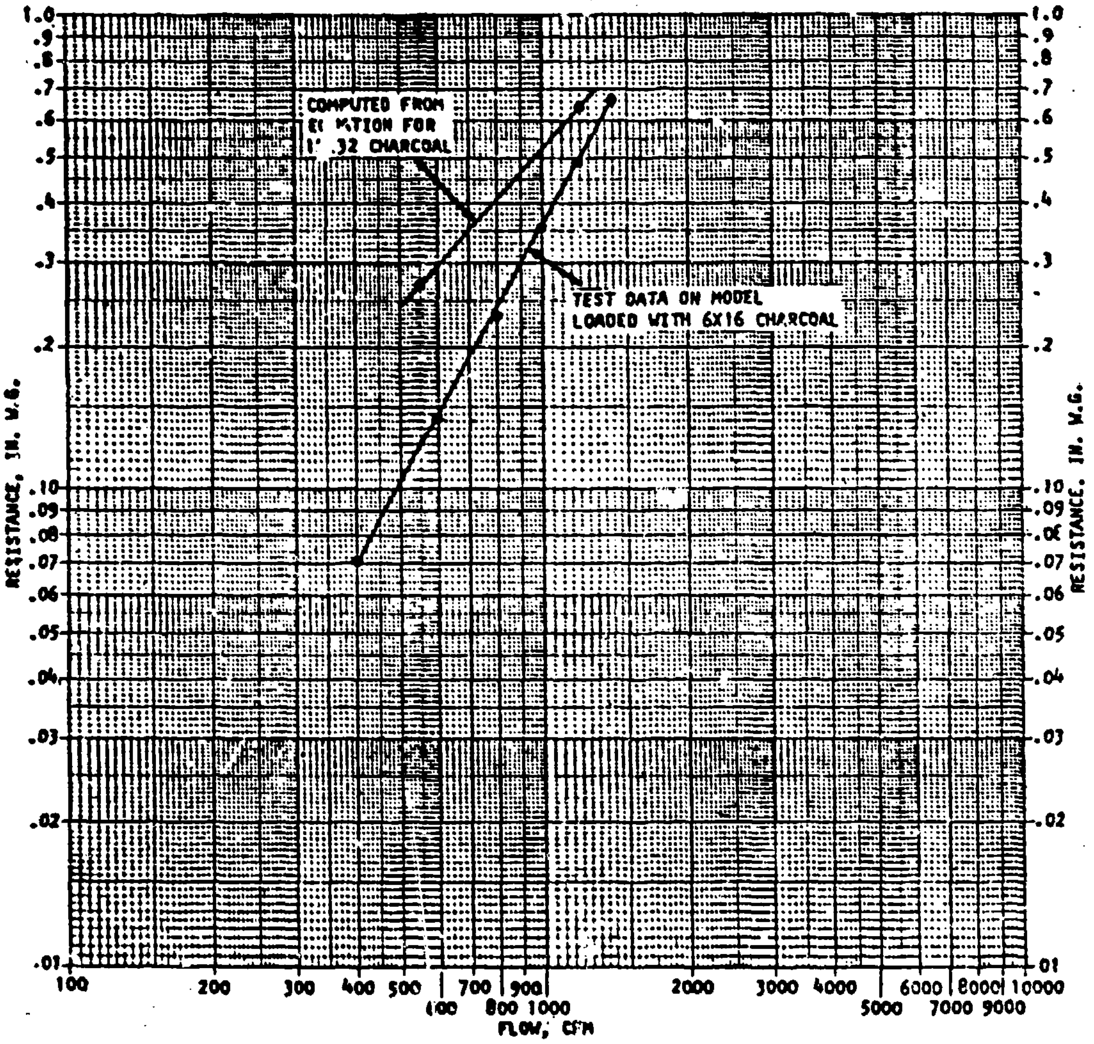


Figure 2-22. Flow Vs. Resistance for Recirculation Filters

E-5

using ASC Whetlerite 12 by 32 mesh charcoal; the design will also be suitable at 1200 cfm using ASC Whetlerite 6 by 16 mesh charcoal.

2-99. PARTICULATE FILTER DESIGN ANALYSIS. In addition to the GPFU filters, two recirculation particulate filter elements are required for the MS1 shelter system, one for the ECU and one for the airlock recirculation unit. The following paragraphs describe the analyses required for selection of the particulate filters

2-100. Characteristics for ECU Filter. The filter for the 1200 cfm ventilation system will use "Biocel" media, which is 95% efficient for the removal of 0.3-micron particles. The available area for the filter is 26 by 16 by 11 inches. A "Biocel" filter of these dimensions will have an effective face area of 2.46 ft^2; the effective face velocity will therefore be 487 fpm. With a media area of 150 ft^2, the media velocity will be 8 fpm.

2-101. Pressure Drop. The pressure drop of this filter may be predicted by the equation:

$$P = 6.22 \times 10^{-8} \left(\frac{\lambda}{m} \left(1 + \frac{z}{h}\right)^2 v_f^2 \right) + K_z \left(1 + \underline{z}\right) \left(\frac{m}{4(1+m)^2} + \frac{m}{(1+m)} \right) v_f$$

Where λ = 0.462 and K = 2.42

P = pressure drop, inches wg

v_f = velocity at filter face, fpm

z = media thickness, inches

h = channel height, inches

m = h/2L, where

2L = length of channel, inches

λ = experimentally determined coefficient, the channel resistance factor, dimensions

K = experimentally determined coefficient of pressure drop across the filter media, inches wg per fpm per inch

Note. This equation is valid for standard air and an absolute viscosity of 0.045 lb/ft/hr. The equation yields a filter pressure drop of 0.83 inches wg. This condition of pressure drop could not be improved by the inclusion of additional media; in fact, additional media will only reduce the channel height, causing an increase in pressure drop.

2-102. A reduction in pressure drop could be obtained by removing media and increasing the channel height. While it is true that the less dense media will result in higher media velocities and thus higher media resistance, the increase in channel height causes a significant reduction in pressure drop compared to an increase in media resistance. A filter with

E-6

only 97 square feet of media will have a pressure drop of 0.75 inches wg. This reduction, however, is not sufficient to trade-off other properties of the filter having a higher pressure drop, i.e., greater structural reliability, longer life, etc.

2-103. The materials of the recommended filter will include glass fiber media, aluminum separators, aluminum frame, fire-retardant neoprene adhesive, and silicone rubber gasket.

2-104. Characteristics For Airlock Filter. The filter used in the airlock recirculation system must have dimensions of 24 by 16 by 11 inches and will be operated at 550 cfm. Made of "Astrocel" material, it will have a media area of 150 square feet and a DOP efficiency of 99.97% (rated efficiency in removing 0.3-micron particles). Its effective face velocity will be 243 fpm; media velocity will be 3.7 fpm. Using the equation in paragraph 2-101, the predicted pressure drop was determined to be 0.7 inches wg. Materials are the same for the airlock filter and for the ECU filter, except that the ECU filter uses glass fiber "Biocel" media.

2-105. ENVIRONMENTAL CONTROL SYSTEM (UNIT)

2-106. The environmental control unit (ECU) contains the necessary components to maintain the interior shelter environment within design parameters. A 60,000 British thermal unit per hour (Btu/hr) heater was selected to provide a minimum temperature of 60°F, dry bulb (db) in ambient temperatures down to minus 25°F. A 3-1/2 ton vapor cycle refrigeration system, using dichlorodifluoromethane refrigerant (R-12), was selected to provide cooling up to an outdoor ambient temperature of 105°F db, 85°F dew point.

2-107. REFRIGERATION SYSTEM DESIGN ANALYSIS. The following paragraphs describe the analyses performed in the design and development of the refrigeration system.

2-108. Load Analysis. The required cooling capacity of the air conditioning system was based on an outside environment of 105°F db (85°F dew point) and a total solar radiation load of 360 Btu/hr/ft^2 at sea level, as defined in AR 705-15, change 1, paragraph 7C, dated 14 October 1963.

2-109. Cooling Load. The cooling load, based upon the design requirements, an outside air intake of 150 cfm, and an interior environment of 90°F db (70% relative humidity) is 39,747 Btu/hr, of which 4,880 Btu/hr is latent heat. The shelter sensible heat load (exclusive of the airlock) is 26,420 Btu/hr; its latent heat load is 2,000 Btu/hr. This results in a room sensible heat ratio of 0.930. It will be shown in the following paragraphs that, from an equipment standpoint, it was not feasible to design the air conditioning system based on a maximum indoor relative humidity of 70%. Instead, the percent of relative humidity must be lowered, increasing the air conditioning latent load on the evaporator.

2-110. Assuming air leaving the evaporator approaches a relative humidity of 95% and removes 26,420 Btu/hr sensible load and a 2,000 Btu/hr latent load from the shelter area to maintain a room condition of 90°F db, 70% relative humidity, the minimum temperature of the shelter supply air would be 79°F db.

E-1